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AN ACCOUNT OF THE COMPARABLE EXPERIMENTS MADE IN THE NEW YORK NAVY YARD, BY CHIEF ENGINEERS THEODORE ZELLER AND GEORGE P. HUNT, OF THE U. S. NAVY, TO ASCERTAIN THE ECONOMIC EFFECT OF USING IN A NON-CONDENSING ENGINE SATURATED STEAM ALONE, AND OF USING IT MIXED WITH COMPRESSED HOT AIR ACCORDING TO THE SYSTEM OF MR. E. M. STRANGE.

BY CHIEF ENGINEER ISHERWOOD, U. S. Navy.

The following experiments were made in the New York Navy Yard on the fifth of August, 1885, by a board composed of Chief Engineers Theodore Zeller and George P. Hunt, of the United States Navy, with an apparatus submitted by Mr. E. M. Strange, for the purpose of ascertaining how much economic gain, if any, attended the use in a non-condensing steam engine of a mixture of saturated steam, and of air compressed to the pressure of the steam, and containing its heat of compression.

The engine to which the apparatus was attached, was an old

one in the boiler shop of the engineering department, and out of use. Its work, when in use, was to drive the shafting from which power was taken to operate the various tools employed in boiler-making, and a small, highly-geared centrifugal blower, which furnished air to the forges of the shop. During the experiments, the tools were not connected with the shafting, and the discharging nozzle of the blower was partly closed, so that the work done by the engine was the forcing of air through the contracted nozzle of the blower, and overcoming the friction of the shafting. Consequently, equal numbers of double strokes made by the piston of the engine in equal times, performed equal work.

The engine and apparatus with which the experiments were made had the following dimensions:

Number of steam cylinders,	1.
Diameter of the steam cylinder,	12 inches.
Diameter of the piston rod on one side of the piston,	1⅜ inches.
Diameter of the piston rod on the other side of the piston,	1¼ inches.
Net area of the piston, exclusive of areas of piston rods,	111·4471 square inches.
Stroke of the piston,	30. inches.
Space displacement of the piston per stroke,	1·9348455 cubic feet.
Diameter of the air compressor,	9. inches.
Diameter of the piston rod of the air compressor (one side of piston only),	1¼ inches.
Net area of the piston of the air compressor, exclusive of area of piston rod,	63·0038 square inches.
Stroke of the piston of the air compressor,	30. inches.
Space displacement of the piston of the air compressor per stroke,	1·093816 cubic feet.
Number of air inlet valves to the air compressor,	1.
Diameter of the air inlet valve,	3. inches.
Number of air outlet valves to the air compressor,	1.
Diameter of the air outlet valve,	3. inches.
Net area of the air outlet and inlet valves, exclusive of stem,	6·96 square inches.
Face of the piston of the air compressor,	3½ inches.
Thickness of metal of the air compressor and of its reservoir,	¾ inch.
Ratio of the space displacement of the piston of the air compressor to that of the piston of the steam cylinder,	0·565325

Ratio of the space displacement of the piston of the steam cylinder to that of the piston of the air compressor, ... 1·768895

The steam cylinder is horizontal and direct acting. It has four valves, two at each end, one of which is for the admission of steam and the other for the exhaust. All the valves are plain plates of metal, and are worked by one eccentric in the ordinary manner. The steam valves act also as cut-off or expansion valves, being tripped by cams at any desired portion of the stroke of the steam piston, and brought to position over the steam port by stiff springs. All the valves work vertically. The steam valves are at the upper side, and the exhaust valves are at the lower side of the cylinder.

The pressure of the steam in the valve chest is upon the backs of the steam valves, and the pressure of the steam in the cylinder is upon the cylinder side of the exhaust valves, while the atmospheric pressure is upon their opposite side. The steam piston is packed by rings and springs in the usual manner.

Previous to commencing the experiments, the piston and valves of the steam cylinder were examined and put in proper working order.

The steam cylinder was neither steam jacketed, nor lagged, nor protected in any way from the radiation of heat. It was situated, however, in a closed room, which prevented air currents from passing over it. The main shaft of the engine carried a light fly wheel.

The air compressor, which was double-acting, was a plain cast iron cylinder placed horizontally behind the steam cylinder, and having its axis in the same straight line with that of the latter. The piston was of cast iron, with two recesses surrounding its face, for the reception of the packing rings and springs which made it air-tight. The piston rod of the air compressor was extended through the back end of the steam cylinder and secured to the piston of the latter, so that, practically, the pistons of the steam cylinder and of the air compressor were on the same rod, and, of course, made stroke for stroke, the piston of the air compressor commencing to compress the air at the same moment that the steam cylinder received steam, both pistons being at the end of their stroke.

By this arrangement, the resistance to the piston of the air compressor increased, while the pressure of the steam upon the piston of the steam cylinder decreased, so that when the two pistons reached the end of their stroke, the resistance against the first was at a maximum, while the pressure upon the last was at a minimum. When the steam was cut off at much less than half the stroke of the piston of the steam cylinder the engine would stop, because the resistance to the piston of the air compressor was then greater than the steam pressure upon the piston of the steam cylinder, so that, notwithstanding the momentum of the fly wheel, the engine gradually came to a state of rest. Of course, this might have been avoided by interposing a lever between the two pistons that would have reversed their movements and made the maximum air compression coincident with the maximum steam pressure upon the piston of the steam cylinder.

Each head of the air compressor was double, the space between the two shells serving as air passages for the incoming and outgoing air. Each end had a receiving and a delivering valve; these valves were single poppet or single disc valves, working horizontally and seating in the inner shell.

The air compressor was concentrically surrounded by a cylindrical reservoir of the same length, and between which and the compressor there was a space of one and one-fourth inches in the clear, which space communicated with the air discharging valves in the ends of the compressor. From the centre of this reservoir, on top, a three-inch interior diameter pipe carried the compressed air to the steam pipe of the engine, debouching into the latter between the throttle valve and the valve chest of the steam cylinder. A short branch from this three-inch diameter air pipe was fitted with a three-inch diameter cock, whereby the air from the compressor was discharged into the atmosphere when the engine was operated by steam alone, the cock being then open. In the three-inch diameter air pipe, and between this branch and the steam pipe, there was a three-inch diameter poppet check valve, whose office was to prevent the steam in the steam pipe from flowing into the atmosphere through the cock when the latter was open.

When the engine was in operation, the air was always being compressed in the compressor and discharged, whether the engine was used with combined steam and air, or with steam alone, so

that in the latter case provision had to be made for its discharge into the atmosphere.

The smallest area for the passage of the compressed air into the steam pipe below the throttle valve, was 3·1416 square inches, being the area due to a diameter of two inches Through this area, equal to one-twentieth of the area of the piston of the compressor, all the compressed air was forced by the latter.

When the engine was in operation with combined air and steam, the pressure in the valve chest of the steam cylinder, in the steam pipe below the throttle valve, in the air pipe from compressor to cylinder, in the reservoir of the air compressor, and in the compressor after the delivery valve opened, was the same, excepting the slight variations in the statical equilibrium made dynamically by the current of the gases.

The experiments were four in number, constituting two pairs of comparable tests with different cylinder pressures, piston speeds and measures of expansion. The largest measure of expansion which could be used was that due to the intended cutting off of the steam at half stroke of the piston. The other measure of expansion was that due to the intended cutting off of the steam at seven-tenths of the stroke of the piston from the commencement.

With each intended measure of expansion, an experiment was first made with combined air and steam, and a certain position of the tripping gear for cutting off the steam. The steam pressure in the boiler was maintained invariable from beginning to end of the experiment; and the throttle valve, having been adjusted to give the intended number of double strokes of piston per minute, remained unaltered during that test. After the experiment commenced, no change of any kind was made in any of the conditions up to its end.

Immediately after the conclusion of the experiment with the combined air and steam, a comparable test was made with steam alone, the position of the cut-off tripping mechanism remaining unchanged from the last experiment, so as to secure as nearly as possible the same measure of expansion for the two cases In this comparable trial, the steam in the boiler was maintained at exactly the same pressure as before, but the throttle valve was altered until the engine piston made the same number of double strokes per minute as before.

During the two experiments with the combined air and steam, indicator diagrams were taken simultaneously every three minutes from both ends of the steam cylinder and from both ends of the air compressor. The indicators remained permanently in place and the same springs were employed throughout. When the experiments were made with steam alone, the indicator diagrams were taken every three minutes from both ends of the steam cylinder only; none being taken from the air compressor because repeated trials showed no sensible pressure therein when the cock in the branch pipe was open to the atmosphere.

Before commencing an experiment, the engine was operated sufficiently long to attain its normal state for the conditions.

The boiler was greatly too large for the engine as loaded, and as the steam in all cases was very much throttled, there was not the slightest trace of foaming or priming. The pressure in the boiler was maintained throughout by a skilled fireman at exactly sixty pounds per square inch above the atmosphere, by opening and closing the furnace door as the pressure tended to rise above or fall below the limit.

A very good spring gauge gave the mean pressure in the valve chest of the steam cylinder.

An excellent thermometer was inserted into the centre of the reservoir of the air compressor, and a duplicate thermometer was inserted into the valve chest of the steam cylinder. These thermometers were screwed into their respective places, so that their naked bulbs and the lower part of their stems were in direct contact with the gases.

The standard Navy Yard barometer gave the pressure of the atmosphere.

The pressures and temperatures were observed and recorded every three minutes.

In commencing an experiment with the combined air and steam, the exact position of the water level in the boiler was marked on the glass water-gauge of the latter, and at the expiration of one hour during which the independent steam pump which fed the boiler was stopped and the feed check valve screwed down, the water level in the boiler was again exactly marked on the gauge. The weight of steam evaporated in the boiler and used in the cylinder during the experiment, was that which was due to the

weight of water between the two levels, the steam pressure being maintained invariable from first to last. The number of cubic feet of water contained between these levels, was ascertained when the boiler was cold by filling it from one to the other by means of a measure of carefully ascertained capacity. The number of pounds of water consumed in any of the cases, was obtained by multiplying the number of cubic feet as above ascertained by the weight of water per cubic foot at the temperature due to the pressure of sixty pounds per square inch above the atmosphere.

The number of double strokes of piston made by the engine, was taken by a counter. The temperature of the air in the engine room, was taken by a thermometer placed near the inlet of the air compressor, but shielded from the radiant heat of surrounding bodies, so as to give the temperature of the air as it entered the compressor.

The data and results of the experiments will be found in the following table, in which the data are the means of all the observations recorded and of all the indicator diagrams taken. For convenience of reference, the columns of the table have been lettered, the quantities grouped and the lines containing them numbered.

(*See Table.*)

The table contains the data and results of the two pairs of experiments made; each pair being the comparable tests with saturated steam alone, and with the same combined with air compressed to the pressure of the steam and retaining the heat of compression.

The columns containing the experimental quantities are lettered *A*, *B*, *C* and *D*. Experiments *B* and *D* were made with saturated steam alone, but with different pressures, measures of expansion, and double strokes of the engine piston per minute. Experiments *A* and *C* were made with saturated steam combined with air compressed to the pressure of the steam and retaining the heat of compression. The intention was that the measure of expansion with which the gases were used, and the number of double strokes made per minute by the piston of the engine, should be exactly the same in experiments *A* and *B* which were to be strictly comparable; and also in experiments *C* and *D* which likewise were to be comparable. This intention was realized as nearly as the nature of the valve gear of the engine would permit, and sufficiently near

to give correct results, the economic differences due to the small variations from the intended conditions being too small sensibly to affect the determinations.

In all the experiments, the steam was used without condensation.

Line 1 gives the time at which each experiment began.

Total Quantities.—Line 2 gives the duration of the experiment in minutes and seconds; and lines 3 and 4 give, respectively, the number of double strokes made by the engine piston, and the number of pounds of feed water consumed, during the time on line 2.

Engine.—Line 5 gives the number of double strokes made by the engine per minute, the quantities being the quotients of the division of those on line 3 by the number of minutes on line 2.

Line 6 gives the steam pressure in the boiler in pounds per square inch above the atmosphere. This was in all cases maintained invariably at sixty pounds.

Line 7 gives the position of the throttle valve as greatly closed in all four experiments. In no two experiments, however, had it the same opening, but even when at the widest, there was still a difference of nearly twenty pounds per square inch between the boiler pressure and the pressure on the steam piston at the commencement of its stroke. The throttle valve was a screw disc and remained unchanged during each experiment.

Line 8 gives the mean pressure in the valve chest of the steam cylinder in pounds per square inch above the atmospheric pressure. The pressure oscillated during each stroke of the steam piston.

Line 9 gives the pressure of the atmosphere in pounds per square inch above zero. This pressure added respectively to the pressures on lines 6 and 8, would give the pressures in the boiler and in the cylinder valve chest above zero.

Line 10 gives the number of pounds of feed water consumed per hour, the quantities being the quotients of the division of those on line 4 by the number of hours on line 2.

Temperatures.—Line 11 gives the mean temperature in the valve chest of the steam cylinder as observed on the thermometer inserted therein. In experiments *B* and *D*, this temperature is that of saturated steam of the pressure on line 8; but in experiments *A* and *C*, it is that which is therein due to the combined steam and air used in them.

Table containing the Data and Results of the Comparable Experiments, made by Chief Engineers Theodore Zeller and George P. Hunt, of the United States Navy, on the 5th of August, 1885, in the New York Navy Yard, for the Purpose of Determining the Relative Economy of Using Saturated Steam alone, and of Using it Mixed with Compressed Hot Air as Produced by the Apparatus of Mr. E. M. Strange and according to his Method.

Number of Line.			Comparable Experiments. The Piston-speed, the Point of Cutting off the Steam, and the Load intended to be the same		Comparable Experiments. The Piston-speed, the Point of Cutting off the Steam, and the Load intended to be the same.	
			Air and Steam Combined. *A.*	Steam Alone. *B.*	Air and Steam Combined. *C.*	Steam Alone. *D.*
1		Time of commencing experiment,	10 50 A. M.	12 12 P. M.	1 56 P. M.	3 13 P. M.
2	Total Quantities.	Duration of experiment in consecutive minutes and seconds,	60 00	59 30	60 00	60 45
3		Total number of double strokes made by the piston of the engine,	2978	2927	3694	3736
4		Total number of pounds of feed water consumed,	776·175452	776·175452	898·880910	898·880910
5	Engine.	Number of double strokes made per minute by the piston of the engine,	49·633333	49·19328	61·566667	61·49794
6		Steam pressure in the boiler in pounds per square inch above the atmosphere,	60	60	60	60
7		Position of the throttle valve,	Greatly closed.	Greatly closed.	Greatly closed.	Greatly closed.
8		Pressure (mean) in the valve chest of the steam cylinder in pounds per square inch above the atmosphere,	24·00	12 25	35·00	20·80
9		Pressure of the atmosphere in pounds per square inch above zero,	14 69	14·69	14 69	14·69
10		Number of pounds of feed water consumed, per hour,	776·175452	782 697935	898·880910	887·783615
11	Temperatures.	Temperature in degrees Fahrenheit in the valve chest of the steam cylinder,	240°	244°	262°	260°
12		Temperature in degrees Fahrenheit due to the pressure in the valve chest of the steam cylinder, supposing the pressure there to be that of saturated steam,	265 1	244·2	280·5	260·0
13		Temperature in degrees Fahrenheit of the air in the reservoir of the compressor,	133·0	. . . .	190·0	. . . .
14		Temperature in degrees Fahrenheit of the air in the engine room,	90·0	94·0	98·0	102·0
15	Steam Pressures in Steam Cylinder.	Fraction completed of the stroke of the piston of the steam cylinder when the steam was cut off,	0·700	0 755	0·512	0 548
16		Fraction completed of the stroke of the piston of the steam cylinder when the steam was cushioned,	0·860	0·885	0·766	0·879
17		Pressure in pounds per square inch above zero in the steam cylinder at the commencement of the stroke of its piston,	41·48	28·64	55·14	36 20
18		Pressure in pounds per square inch above zero in the steam cylinder at the point of cutting off the steam,	30·32	21·05	39·39	24·87
19		Pressure in pounds per square inch above zero in the steam cylinder at the end of the stroke of its piston,	22·08	16·50	22·11	14·02
20		Pressure in pounds per square inch above the atmosphere in the steam cylinder of the back pressure against the piston, exclusive of the cushioned back pressure,	0 50	0·50	0·50	0·50
21		Mean indicated pressure on the piston of the steam cylinder in pounds, per square inch,	15·90	7·05	21·00	8·15
22		Mean total pressure on the piston of the steam cylinder in pounds, per square inch,	28·96	20·49	34·15	21·50
23		Mean pressure of the expanding steam in the steam cylinder in pounds per square inch above zero, after the cut-off valve closed,	22·90	16·90	25·87	17·00
24	Air Pressures in Air Compressor.	Fraction completed of the stroke of the piston of the air compressor when its delivery valve opened,	0·533	. . . .	0·419	. . . .
25		Pressure in pounds per square inch above the atmosphere in the air compressor at the commencement of the stroke of its piston,	0·00	. . . .	0·00	. . . .
26		Pressure in pounds per square inch above the atmosphere in the air compressor when its delivery valve opened,	20·00	. . . .	32·49	. . . .
27		Pressure in pounds per square inch above the atmosphere in the air compressor at the end of the stroke of its piston,	28·70	. . . .	45·09	. . . .
28		Mean indicated pressure on the piston of the air compressor in pounds per square inch,	15·49	. . . .	22·69	. . .
29		Mean pressure of the air during compression in the air compressor in pounds per square inch above the atmosphere; that is to say, the mean pressure of the compressed air between the commencement of the stroke of the piston of the air compressor and the point at which its delivery valve opened,	7·712	. . . .	11·203	. . . .
30	Horses-power.	Indicated horses-power developed by the steam cylinder,	13·325865	5·836252	21·831810	8·463364
31		Total horses-power developed by the steam cylinder,	24 971512	17·020510	35·502681	22·326665
32		Indicated horses-power expended upon the air in the air compressor,	7·339162	. . . .	13·335262	. . . .
33		Horses-power expended in compressing the air in the air compressor; that is to say, the horses-power developed in compressing the air in the air compressor from the commencement of the stroke of its piston to the point at which its delivery valve opened,	1·947553	. . . .	2·758770	. . . .
34		Indicated horses-power developed by the steam cylinder less the indicated horses-power expended upon the air in the air compressor,	5·986703	. . . .	8·496548	. . . .
35		Total horses-power developed by the steam cylinder less the indicated horses-power expended upon the air in the air compressor,	16 932350	. . . .	22·167419	. . . .
36		Horses-power developed by the expanding steam in the steam cylinder, after the closing of the cut-off,	5·757779	3·439406	13 124619	7·979446
37	Economic Results.	Number of pounds of feed water consumed per hour per indicated horses-power developed by the steam cylinder less the indicated horses-power expended upon the air in the air compressor,	129·666605	133 631683	105·793660	104 897251
38		Number of pounds of feed water consumed per hour per total horse-power developed by the steam cylinder less the indicated horses-power expended upon the air in the air compressor,	45·839795	45 985574	40·549642	39 763378
39	Weight of Steam accounted for by the Indicator.	Number of pounds of steam present in the steam cylinder at the point of cutting off the steam, calculated from the pressure there,	. . . .	459·471009	. . . .	499·684 50
40		Number of pounds of steam present in the steam cylinder at the end of the stroke of its piston, calculated from the pressure there,	. . . .	481 231508	. . . .	524·730053
41		Number of pounds of steam condensed in the steam cylinder to furnish the heat transmuted into the total horses-power developed by the expanded steam alone,	. . . .	8 981397	. . . .	21·102064
42		Sum of the two immediately preceding quantities,	. . . .	490·212905	. . . .	545·832117
43	Difference between the Weight of Water vaporized in the Boiler and the Weight of Steam accounted for by the Indicator.	Difference, in pounds, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the steam cylinder at the point of cutting off the steam,	. . . .	316·704443	. . . .	399·196860
44		Difference, in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the steam cylinder at the point of cutting off the steam,	. . . .	40·803203	. . . .	44·410428
45		Difference, in pounds, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the steam cylinder at the end of the stroke of its piston,	. . . .	285·962547	. . . .	353·048793
46		Difference, in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the steam cylinder at the end of the stroke of its piston,	. . . .	36 842514	. . . .	39·276481

Line 12 gives the temperature, according to Regnault, due to the pressure in the valve chest of the steam cylinder, supposing that pressure to be the pressure of saturated steam. In experiments *B* and *D*, in which saturated steam was used, the agreement between the observed temperature and Regnault's is exact, showing the accuracy of the instruments employed.

In experiments *A* and *C*, in which the mixture of steam and air was used, the pressure of the steam component in the valve chest is assumed to be that on line 8, and the temperature given on line 12 is Regnault's temperature for that pressure. This assumption supposes there was no mixing of the air and steam in the valve chest, and that the air delivered into the chest from the compressor remained there *en bloc*, the steam entering from the boiler occupying the remainder of the space *en bloc*. While the air was being delivered into the valve chest by the compressor, it excluded the entrance of steam therein, because the pressure of the entering air must, in order to enter, be necessarily greater than the pressure of the steam already within, the steam entering only after the entire charge of air had been received. On this assumption, the engine, during experiments *A* and *C*, was not operated by a *mixture* of steam and compressed air, but by steam and compressed air *successively* and *separately*, each of these fluids remaining distinct from the other and performing its work in succession. Hence, the observed temperature in the valve chest of the steam cylinder (line 11) should be determined by the temperature of the steam therein (line 12), and by the temperature of the compressed air (line 13), multiplied by their respective *times of occupation* of the valve chest, and the sum of the products divided by the sum of the times. A rough calculation will show that the observed temperature on line 12 is just what would be given by such a calculation.

When the facts are considered, that the air compressor delivers its compressed air into the valve chest against the valve chest pressure without interruption, excluding *en bloc* whatever opposes the incoming air, and that only after the mass of air is withdrawn by the steam cylinder from the valve chest can steam enter the latter from the boiler, and that the time any single charge of air and steam remains in the valve chest is extremely limited (0·6 second in experiment *A* and 0·5 second in experiment *C*), the

very small chance for any mixture of importance can be easily comprehended; and as the same conditions exist for the cylinder in these respects as for the valve chest, there is just as little chance for mixture there.

These experiments, therefore, were not, and could not be made under the conditions necessary to show whether any economy of steam could be effected by the use of *intimately mixed* air and steam. The *mixing*, absolutely essential and by which alone any economic gain could be accomplished, was entirely wanting; and how any mixing is to be produced, unless by bulky and inconvenient mechanical methods, is difficult to conceive.

If the valve chest contained a thorough mixture of the air and steam, then the total pressure therein (line 8) would be composed in part of the steam pressure and in part of the air pressure, in proportion to their respective bulks at that pressure; so that the steam pressure would be much less than the pressure on line 8, and the air pressure would also be much less, the sum of the two pressures making up the total on line 8; whereas, without mixing, both the steam and the air have the pressure on line 8.

With a thorough mixture of the air and steam, the temperature in the valve chest would be very different from that on line 11, and would be determined by the respective weights of air and steam multiplied by their respective temperatures, and by their respective specific heats per unit of weight, the sum of the final products being divided by the aggregate weight of the air and steam, the quotient would be the temperature in the valve chest.

Line 13 gives the mean temperature of the air in the reservoir of the compressor, and line 14 gives the temperature of the air in the engine room near the receiving valves of the compressor. The temperature in the reservoir is that which is due to the heat of compression less the initial temperature, and, all other things being the same, is considerably affected by the more or less aqueous vapor in the air, the more the vapor the lower the temperature. Of course, the heat of compression, other things equal, results from and is in proportion to the mechanical work done upon the air during its compression.

Steam Pressures in the Steam Cylinder.—All the quantities on lines 15 to 23, both inclusive, are the means from all the indicator diagrams taken from the steam cylinder. Line 15 gives the fraction completed of the stroke of the steam piston when the steam

was cut off. The point of cutting off was taken to be that at which the convex curve of the throttled steam reversed into the concave curve of the expanded steam.

Line 16 gives the fraction completed of the return or exhaust stroke of the piston, when the exhaust port of the cylinder was closed and the cushioning or compression of the back pressure steam in the cylinder commenced. This compression was sufficient in all cases to fill the clearance and steam passage at one end of the cylinder to the valve chest pressure, so that the steam drawn from the boiler per stroke of piston was unaffected by those spaces.

Lines 17, 18 and 19 contain respectively the steam pressures in the cylinder in pounds per square inch above zero, at the commencement of the stroke of the piston, at the point of cutting off the steam, and at the end of the stroke of piston.

Line 20 contains the pressure of the back pressure against the piston, exclusive of the cushioned back pressure, in pounds per square inch above the atmosphere. The back pressure on line 20 operated against the piston for the fraction of its stroke given on line 16.

Line 21 gives the mean indicated pressure on the steam piston in pounds per square inch of its area.

Line 22 gives the mean total pressure on the steam piston above zero, in pounds per square inch of its area. If the quantity on line 20 be added to the atmospheric pressure on line 9, and the sum reduced in the ratio of the fraction on line 16 to unity, and added to the quantity on line 21, the sum will be the quantity on line 22.

Line 23 gives the mean pressure of the expanding steam in the cylinder in pounds per square inch above zero. That is to say, the mean pressure of the steam above zero for the portion of the piston stroke after the closing of the cut-off valve.

Air Pressures in the Air Compressor.—Lines 24 to 29, both inclusive, contain the air pressures in the air compressor as given by the mean of all the indicator diagrams taken from the compressor. There must be remembered that the air and steam pistons being upon the same piston rod, the compression of the air commenced simultaneously with the commencement of the stroke of the steam piston, and attained its maximum when the stroke of

that piston ended, so that the maximum air compression and the minimum steam pressure were simultaneous.

Line 24 gives the fraction of the stroke of the air piston completed when the delivery valve of the compressor opened and the compressed air commenced flowing into the reservoir of the compressor. This valve, of course, opened when the pressure in the compressor exceeded the pressure in the reservoir.

Line 25 gives the pressure in pounds per square inch above the atmosphere of the air in the compressor when the air piston commenced its stroke. This pressure was exactly the atmospheric pressure (line 9).

Lines 26 and 27 give, respectively, the air pressures in the compressor in pounds per square inch above the atmosphere, at the point in the stroke of the air piston at which the delivery valve opened, and at the end of the stroke of the piston. It will be observed that the latter pressure is considerably greater than the former, and is owing to the fact of a corresponding difference in the pressure in the reservoir at these times. This difference is caused by the variability in the draught upon the reservoir made by the piston of the steam cylinder, and its effect must be included in estimating the heat of compression.

Line 28 gives the mean indicated pressure in pounds per square inch of the air in the compressor upon its piston.

Line 29 gives the mean pressure in pounds per square inch above the atmosphere of the air in the compressor, between the commencement of the stroke of its piston and the point at which the delivery valve opened.

Horses-power.—Lines 30 to 36 contain the various horses-power developed by the piston of the steam cylinder, and also the horses-power expended upon the piston of the air compressor.

Line 30 gives the number of indicated horses-power developed by the steam cylinder, calculated for the pressure on line 21, and consequently inclusive in the cases of experiments *A* and *C* of the indicated horses-power expended upon the air in the air compressor as given on line 32.

Line 31 gives the number of total horses-power developed by the steam cylinder, calculated for the pressure on line 22, and consequently inclusive in the cases of experiments *A* and *C* of the indicated horses-power expended upon the air in the air compressor as given on line 32.

Line 32 gives the number of indicated horses-power expended upon the air in the air compressor, calculated for the pressure as given on line 28.

Line 33 gives the number of horses-power expended in compressing the air in the air compressor from the commencement of the stroke of its piston to the point in that stroke at which the delivery valve opened, calculated for the pressure on line 29. These horses-power are not the entire horses-power expended in air compression, because, after the delivery valve of the air compressor opened, there was a further compression of the air until the end of the stroke of the piston was reached, owing to the fact that the pressure in the reservoir into which the delivery valve opened was not constant, but was greater when the piston was at the end of its stroke than when it was at the point in its stroke at which the delivery valve opened. The temperature of the compressed air in the reservoir of the compressor was that which is due to the entire work of compression done upon the air.

Line 34 gives the indicated horses-power developed in the steam cylinder after subtracting the indicated horses-power expended upon the air in the compressor. The remainder is the true indicated horses-power developed by the steam cylinder during experiments *A* and *C*, and is obtained by subtracting the quantities on line 32 from those on line 30.

Line 35 gives the total horses-power developed in the steam cylinder after subtracting the indicated horses-power expended upon the air in the compressor. The remainder is the true total horses-power developed by the steam cylinder during experiments *A* and *C*, and is obtained by subtracting the quantities on line 32 from those on line 31.

Line 36 gives the number of horses-power developed in the steam cylinder by the expanding steam after the closing of the cut-off valve, calculated for the pressure on line 23.

Economic Results.—Line 37 gives the cost of the indicated horse-power developed by the steam cylinder, in pounds of feed-water consumed per hour. For experiments *A* and *C*, these quantities are the quotients of the division of the quantities on line 10 by those on line 34. For experiments *B* and *D*, these quantities are the quotients of the division of the quantities on line 10 by those on line 30.

Line 38 gives the cost of the total horse-power developed by the steam cylinder, in pounds of feed-water consumed per hour. For experiments *A* and *C*, these quantities are the quotients of the division of the quantities on line 10 by those on line 35. For experiments *B* and *D*, these quantities are the quotients of the division of the quantities on line 10 by those on line 31.

The total horses-power represent the entire work done by the steam. They include the indicated horses-power and the horses-power required to push out of the steam cylinder the back pressure against its piston; consequently, the total horse-power is the proper unit whose cost in heat, represented by the weight of feed water consumed per hour, is to be used as the correct economic result for comparison in cases like those of the present experiments.

Weight of Steam accounted for by the Indicator.—The weight of steam accounted for by the indicator, is the sum of the weight of steam present as such in the cylinder at any given point of the stroke of its piston, and of the weight of steam which had been condensed to furnish the heat transmuted into the power developed by the expanding steam at that point, calculated for the horses-power developed between the closing of the cut-off valve and the point in question, according to the indicator. If the point taken in the stroke of the piston is previous to, or at the point of cut-off, then there is no such condensation of the steam to be calculated, and the weight determined by the pressure shown by the indicator is all there is to be considered.

Line 39 gives the number of pounds of steam present in the steam cylinder at the point of cutting off during the experiments, calculated from the pressure on line 18. As the clearance and steam passage at one end of the cylinder were filled by the compressed back pressure due to the cushioning, these spaces are not included in the calculation.

Line 40 gives the number of pounds of steam present in the steam cylinder at the end of the stroke of its piston during the experiments, calculated from the pressure on line 19. For the reason just stated, the spaces in the clearance and steam passage are not included.

Line 41 gives the weight of steam condensed during the experiments, in the cylinder to furnish the heat transmuted into the

horses-power (line 36) developed by the expanding steam between the point of cutting off and the end of the stroke of the piston. In making this calculation, the dynamical value of one Fahrenheit unit of heat is taken at 789¼ foot-pounds, which makes the thermal equivalent of one horse-power $\left(\frac{33{,}000}{789\frac{1}{4}} = \right)$ 41·811847 Fahrenheit units; consequently, the number of horses-power on line 36 multiplied by 41·811847, and by the duration of the experiment in minutes, and the product divided by the latent heat due to the pressure on line 23, gives the quantities on line 41.

On line 42 is the sum of the quantities on lines 40 and 41, this sum being the number of pounds of steam accounted for during the experiments, by the indicator at the end of the stroke of the steam piston.

It will be observed that the weight of steam in the cylinder during experiments *A* and *C*, in which steam and air were used together, cannot be determined by the indicator, because the respective bulks of those substances present at the point of cutting off, and at the end of the stroke of the piston, cannot be known.

Difference between the Weight of Water Vaporized in the Boiler and the Weight of Steam accounted for by the Indicator.—Line 43 gives the difference between the quantities on line 4 and on line 39, and shows the number of pounds of steam condensed in the cylinder, during the experiments, by the interaction of its metal, at the point of cutting off.

Line 44 shows the per centum which the quantities on line 43 are of the quantities on line 4.

Line 45 gives the difference between the quantities on line 4 and on line 42, and shows the number of pounds of steam condensed in the cylinder during the experiments by the interaction of its metal, at the end of the stroke of its piston.

Line 46 shows the per centum which the quantities on line 45 are of the quantities on line 4.

An examination of the economic results (line 38) given in the case of the two comparable experiments *A* and *B*, shows that the total horse-power developed by the engine was obtained for exactly the same weight of feed-water consumed per hour, whether steam was used alone or in combination with air.

A similar examination in the case of the two comparable experiments *C* and *D*, shows precisely the same result.

Hence, *under the conditions of the experiments,* there was neither gain nor loss in economy of fuel due to the use of combined air and steam, comparably with the use of steam alone.

The question: On what grounds could any other result be expected? may be anticipated here and a reply given.

In the working of surface condensing engines, particularly of those in which steam of very low pressure was used with a high measure of expansion and an excellent vacuum (from twenty-six and one-half to twenty-seven and one-half inches of mercury), the observation has been made that oftentimes the condensation of the steam in the cylinder was so great as to produce a slapping noise whenever the piston came to the end of its stroke, and that the cylinder relief valves in this case had to be in constant use for the discharge of this water of condensation; but when enough air was allowed to leak into the condenser to reduce the vacuum to twenty-three or twenty-four inches of mercury, this cylinder condensation was so much decreased that the slapping noise ceased and the relief valves needed not to be used.

When the engine has a *closed* hot well or air pump reservoir, most of the air discharged from the condenser by the air pump is, instead of being thrown into the atmosphere, forced by the feed pump into the boiler, together with the feed water, the vacuum being thereby proportionally reduced, as this air returns to the condenser increased by whatever more may have leaked in at the various vacuum joints of the engine. Of course, after the air had accumulated in the reservoir beyond the capacity of the feed pump to draw off below the atmospheric pressure, the automatic discharge valve of the reservoir opened and delivered the air into the atmosphere, after which the discharge valve closed, and the same cycle was repeated.

Most feed-pumps are fitted with air-chambers to cushion the water discharge upon the contained air, thereby relieving the mechanism from shock and unnecessary strain, and producing a continuous flow to the boiler notwithstanding the intermitting action of the pump. Now, if a cock be placed on the top of the air chamber, it will, when left open, discharge continuously a considerable quantity of air, producing a corresponding improvement of the vacuum over what the latter was with the cock closed. Of course, the opening of the cock must be so graduated that only

the air will be discharged—no water—which is easily effected after a few trials. The squealing of the escaping air is a nuisance, but there is no other objection to the method. Cases have been known in which the vacuum has been thus improved several inches of mercury, but always with increased cylinder condensation, as evidenced by the production of the slapping noise in the cylinder caused by the increase of the water of condensation, due to the less air in the steam. In some cases, no increase in the economy of fuel followed the improved vacuum—the engine doing the same work—the loss by the greater cylinder condensation equalling the gain by the better vacuum, the temperature of the feed water remaining constant.

Steam, when intimately mixed with air, will not liquefy at the temperature that will condense it when unmixed. That the condensation of steam is greatly lessened by the admixture of air with it, is shown upon a large scale in the atmosphere which holds aqueous vapor in the state of gas at temperatures enormously below the liquefaction point of the vapor when not mixed with air, the vapor in such cases being superheated, and the greater the proportion of air, the less condensable is the vapor, other things equal.

The only direct experiments upon the influence of air mixed with steam in preventing the condensation of the latter upon metallic surfaces colder than itself, are those made in 1873, by Professor Osborne Reynolds, M. A., Fellow of Queen's College, Cambridge, England, a *résumé* of which was published in the *London, Edinburgh and Dublin Philosophical Magazine*, for 1874, which *résumé* is as follows, except that the description of the apparatus used and of the mode of experimenting with it are omitted:

"The object of this investigation is to ascertain how far the presence of a small quantity of air affects the power of a cold surface to condense steam. *A priori*, it seemed probable that it might retard condensation very much; for when pure steam comes up to a cold surface and is condensed, it leaves an empty space, which is immediately filled with fresh steam; so that the passage of the steam up to the cold surface is unobstructed, and if the surface could carry off the heat fast enough, then the rate of condensation would be unlimited. If, however, the steam is mixed with

air, then, as the mixture comes into contact with the cold surface, the steam will be condensed and the air will be left between the fresh steam and the cold surface; so that after condensation has commenced that surface will be protected by a stratum of air, and fresh steam will have either to displace this or pass through it before it in turn can be condensed.

"This question, besides its philosophical interest, has important practical bearings on the steam engine.

"(1.) If the quantity of air mixed with the steam affects the rate at which it condenses, then the ratio which the pressure of air bears to the pressure of steam in a condenser will materially affect its efficiency; this is particularly important with reference to the *surface condenser.*

"(2.) If air prevents [retards] the condensation of steam, then by sending air into the boiler of a *high pressure engine* [non-condensing engine] the condensation at the surface of the cylinder will be prevented [lessened] which, if allowed to occur, becomes a source of great waste; for when the steam comes into a cold cylinder it condenses, heating the cylinder and leaving water, which will again be evaporated as soon as the steam escapes; and this, in evaporating, will cool the cylinder. By preventing this, the mixing of air with the steam would effect the same object as the steam-jacket, only in a more efficient manner; for the heat communicated to the steam in the cylinder from the jacket, is not nearly so effective as that which is communicated from the boiler, in consequence of the steam in the cylinder being at a lower temperature than in the boiler.

"In making these experiments two objects were particularly kept in view.

"(1.) To ascertain if there is a great difference in the rate of condensation of pure steam and a mixture of steam and air—to ascertain in fact whether pure steam condenses at an unlimited speed.

"(2.) To ascertain if (and according to what law) the effect of air on the condensation increases as the proportion of air to steam increases.

"Of these two undertakings the first is the most difficult. The rate of condensation of pure steam is so great that it is practically impossible to measure it; and to institute a comparison between this and the condensation of a mixture of steam and air is like

comparing the infinite with the finite. It is practically impossible to keep any surface cold when an unlimited supply of pure steam is condensed upon it, so that under such circumstances the quantity of pure steam condensed is limited by the power of the surface to carry off the heat."

* * * * * * * * * * * *

"*Conclusions.*—The conclusions to be drawn from these experiments are as follows:

"(1.) That a small quantity of air in steam does very much retard its condensation upon a cold surface; that, in fact, there is no limit to the rate at which pure steam will condense but the power of the surface to carry off the heat.

"(2.) That the rate of condensation diminishes rapidly and nearly uniformly as the pressure of air increases from two to ten per centum that of the steam, and then less and less rapidly until thirty per centum is reached, after which the rate of condensation remains nearly constant.

"(3.) That in consequence of this effect of air the necessary size of a surface condenser for a steam engine increases very rapidly with the quantity of air allowed to be present within it.

"(4.) That by mixing air with the steam before it is used, the condensation at the surface of a cylinder may be greatly diminished, and consequently the efficiency of the engine increased.

"(5.) That the maximum effect, or nearly so, will be obtained when the pressure of the air is one-tenth that of the steam, or when about two cubic feet of air at the pressure of the atmosphere and the temperature 60° F. are mixed with each pound of steam.

"*Remarks.*—As the investigation was nearly completed, my attention was called to a statement by Sir W. Armstrong, to the effect that Mr. Siemens had suggested as an explanation of the otherwise anomalous advantage of forcing air into the boiler of a steam engine, that the air may prevent, in a great measure, the condensation at the surface of the cylinder. It would thus seem that Mr. Siemens has already suggested the probability of the fact which is proved in this investigation. I am not aware, however, that any previous experiments have been made on the subject, and therefore I offer these results as independent testimony of the correctness of Mr. Siemens's views as well as of my own."

The most potent cause of the lessened condensation of steam when mixed with air, is the separation of the molecules of the former by the molecules of the latter; for, evidently, there can be no condensation unless the steam molecules can coalesce; and, further, the steam when thus mixed with air is in the superheated state; that is, it has a temperature higher than is normal to its pressure as saturated steam, and as that excess of temperature must be removed before liquefaction can commence, the rate of condensation must be proportionally lessened.

When air is forced into a boiler or valve chest of a steam engine against a pressure greater than that of the atmosphere, it enters with the heat of compression, the temperature due to which may be greater than the temperature of the steam therein, and this higher temperature becomes a source of still greater superheating of the steam.

Professor Reynolds makes no allusion to these causes, and ascribes the entire retardation of condensation due to the air admixture, to the air preventing the steam from gaining access to the surface, blocking it out from reaching the surface so to speak. The final effect in the present case, as in all physical phenomena, is doubtless due to several causes, the one cited by the Professor being probably the least.

The surfaces of a cylinder or condenser when in use, are always covered with a stratum of more or less tenuity of water of condensation which, in its turn, is always covered with a stratum of more or less tenuity of air, and it is upon this air coating that any new discharge of steam will be delivered and not directly upon the metallic surfaces. The condensing power of these surfaces must be exercised upon the steam across both strata of water and air. If the Professor's reasoning were correct as to the retardation of condensation by the air molecules excluding the steam molecules from contact with the metallic surfaces by interposition, there should be no condensation at all, for the latter certainly never come in direct contact with them.

The use of air in connection with steam will not produce any economic effect, unless the two are *intimately* mixed; merely delivering them in masses together avails nothing. They must not only be mixed but *thoroughly* mixed, the molecules of the one must be placed between the molecules of the other, and there is, of

course, great difficulty in producing any such interposition. Time for what may be called natural mixing is utterly wanting during the operation of a steam engine, and if the mixture is to have its full effect in this case, some artificial means will have to be devised to produce it.

In the experiment with Mr. Strange's apparatus, the steam and the air were delivered into the cylinder in distinct masses, separately and successively. There was no time for natural mixing, and no provision had been made for artificial mixing by means of appropriate mechanism, consequently, no saving of fuel was produced; but this want of economic effect was not due to any unsoundness or error in the principle or theory, but solely to the fact that the principle was not applied under the conditions requisite for success. And herein will be found the explanation of many similar failures in which the practical results were a most disappointing outcome from unquestionably sound premises, the premises had not been realized and the failure was in that fact and not in them.

The difficulties in the way of practically realizing the idea of using a mixture of steam and air in a steam engine, are very considerable, not only must some artificial means of producing the proper mixing be devised, but the temperature due to the heat of the air compression in the compressor must be kept below the point at which the metals of the compressor will not work. A water-jacket around the compressor would doubtless accomplish this, but such an expedient involves pumps, extra water supply, pipes, etc., all of which are extremely objectionable as well as costly, and only to be resorted to when the gain is very considerable.

Some thirty or forty years ago, a Mr. Storms produced what he called "the Cloud engine," which was a non-condensing steam engine operated by a mixture of air and steam. In this case the air pump or compressor delivered the air at large into the steam room of the boiler, depending upon the conditions existing therein to effect the proper mixing. An experiment was made with this system on a very small steam cylinder at the "Novelty Iron Works," New York City, the power in the two cases of using first the mixture and then the steam alone being measured by a friction brake. The writer has no knowledge of the details of the

apparatus, nor of the experiment which was said to have been satisfactory in showing a very large economy in favor of the mixture. Chief Engineers Zeller and Hunt would have repeated this experiment by means of Mr. Strange's apparatus, delivering the compressed air from the compressor into the steam room of the boiler instead of into the valve-chest of the cylinder, had they not been prevented by a decision of the Navy Department not to allow the least expenditure of money for experimental purposes in steam engineering, the cost of making Mr. Strange's apparatus and experimenting with it having been entirely defrayed by that gentleman.

In the case of all steam engines, there is a very considerable though unintentional mixture of air with the steam. The feed pumps force air as well as water into the boilers, but in unknown proportions. This air, on arriving in the boiler, is immediately disengaged from the water, rises into the steam room, and passes with the steam in a more or less mixed condition to the cylinder, and thence to the condenser, lessening, without doubt, the condensation of steam in the first, due to the interaction of its metal, and in the latter by the presence of the mixed air, with the result that less fuel is required to produce a given engine-power as measured in the cylinder by an indicator (not by a friction brake), and more of this power is required to work the feed pumps and to pump out the air from the condenser, the vacuum in the latter and the temperature of the feed water being maintained constant in both cases. Also, a larger condenser and air pump would be required to produce the same condensation and vacuum; but a smaller boiler would supply the steam for the given power, the rate of combustion of the same fuel remaining the same. It is something, however, for engineers to know that the inleakage of air is not so entirely detrimental as is usually supposed, and that it has some compensations for its obvious disadvantages.

Supposing, now, that a method of perfectly mixing the air and steam were devised, then, as the beneficial effects of using this mixture consist entirely in the lessened cylinder condensation due to the interaction of its metal, the benefit would be diminished exactly in the proportion that cylinder condensation is diminished by other causes. Thus, other things equal, the larger the cylinder, the thinner its metal; the nearer the proportion of its diameter to its

length for producing the minimum surface with a given space displacement of piston per stroke, the less surface in the steam passages; the application of a steam jacket to the cylinder, the lower the initial pressure of the steam and the higher the back pressure against the piston, the smaller the measure of expansion with which the steam is used, the more the steam is superheated; the greater the reciprocating speed of the piston, etc., etc., the less will be the economic effect obtained from the admixture of air.

That the intermixing of air with steam would prevent the condensation of the latter, is by no means a recent discovery; it was very early known in the use of steam for mechanical purposes, having been practiced certainly not later than 1766 in Savery's apparatus, as improved by Desagulier and others for raising water by means of the vacuum formed by the condensation of steam; and it also constituted a feature of Newcomen's atmospheric engine about 1772, after the latter had been perfected by the celebrated Smeaton. Farey describing this engine, in 1827, says:

"It was found advantageous to admit a continual leakage of air into the cylinder; for this purpose a small cock was inserted into the bottom of the cylinder, and was opened as much as it could be to allow the piston to complete its working stroke. When this cock was shut, the engine required a greater counterweight to return the piston, or else it would move slower in the returning stroke, so that in either case the power of the engine was lessened, when it worked without this leakage of air; and the effect of a given quantity of fuel was lessened by shutting the air-cock, in the proportion of twelve to eleven, and in other cases as twelve to ten.

"The air thus admitted into the cylinder, would have a beneficial effect to diminish the condensation of the steam against the internal surface of the cylinder, because the air would have a tendency to collect against that surface, and form a non-conducting lining to the cylinder, so as to keep the steam from actual contact with it.

"As the advantage of admitting air into the cylinder is only to diminish condensation, it is evident that in those engines which condense the greatest proportion of steam, the advantage of air will be greatest."

The use of air mixed with steam in any considerable proportion is probably confined to non-vacuum engines; the number of these,

however, is very great and their consumption of fuel enormous. In view, then, of the possible economic value of the problem to the world, an early effort should be made to exhaustively solve it, for the importance of any method that will lessen—though ever so little—the cost of fuel for the great motor of civilization on which the progress of our race seems to depend, is so great that the expense of the solution appears comparably as nothing.

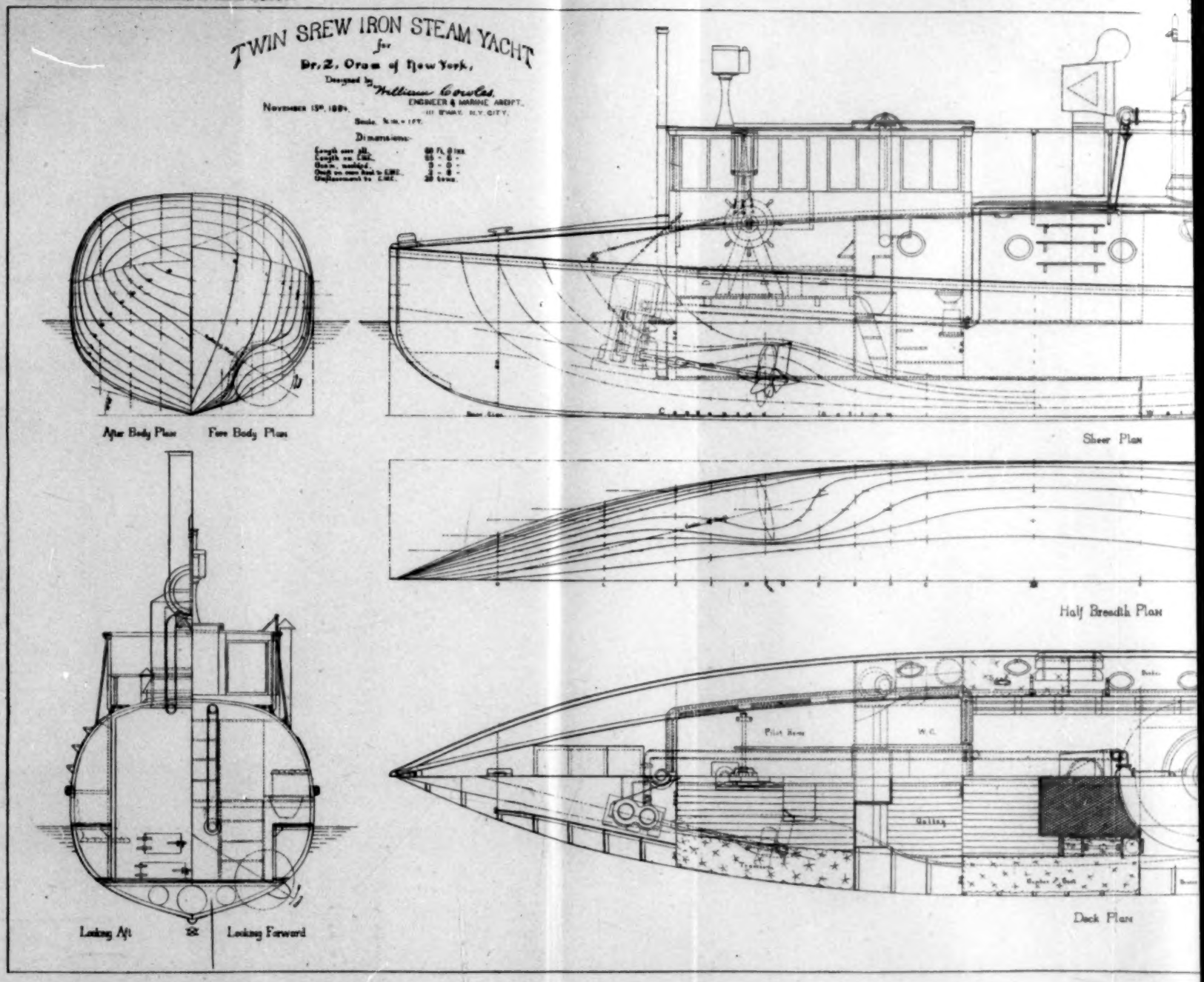
If an experiment be made with mixed air and steam, by delivering the compressed hot air into the steam room of a boiler, the air should be delivered first into perforated pipes arranged in gridiron form over the entire area of the steam room; and it should then emerge through the small holes of these pipes into the steam room, thereby obtaining a more intimate mixture with the steam in the short time available for that purpose, than if delivered into the steam room in bulk.

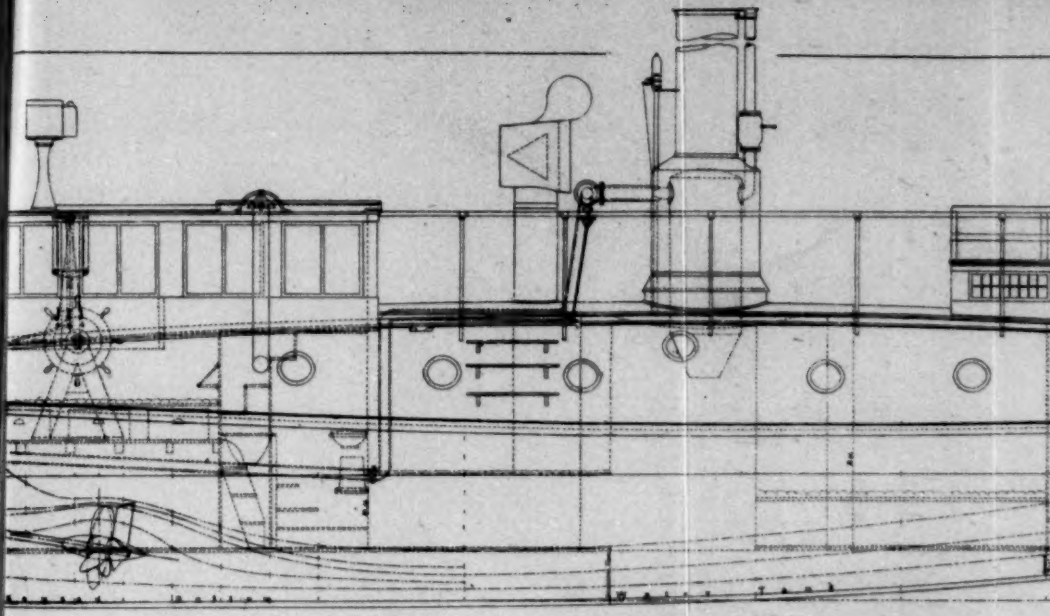
The subject is of sufficient importance and promise to warrant an exhaustive experimental investigation, and one of the objects of this paper is, by supplying the facts thus far obtained, to guide the future experiments by the shortest and surest path.

QUANTITY OF ELECTRICAL ELEMENTARY PARTICLES.—Herwig attempted to estimate the magnitude of electrical elementary particles, but he made some very arbitrary assumptions, and confounded the ideas of "mass" and "quantity" of particles. E. Budde proposes the questions: (1.) Are there in nature discrete elementary particles of electricity? (2.) What is their magnitude? According to the analogy of conclusions drawn for carbon, the questions may be answered as follows: If there are in nature discrete elementary particles of electricity, it is to be expected that an absolutely definite, very small quantity of electricity occurs, and plays an important part in a large number of processes. If experiment shows us such an amount, then that amount of electricity is the probable quantity of the particles of electricity. The region in which we must investigate, is that of those processes in which electricity interacts with ponderable atoms, and defines the action of those ponderable atoms; that is to say, the region of electrolytical decompositions and combinations. From considerations based upon Faraday's law, he deduces the value

$$E = 0{\cdot}00000051 \text{ mg.} \tfrac{1}{2} \text{ mm.} \tfrac{3}{4} \text{ sec.} - 1.$$

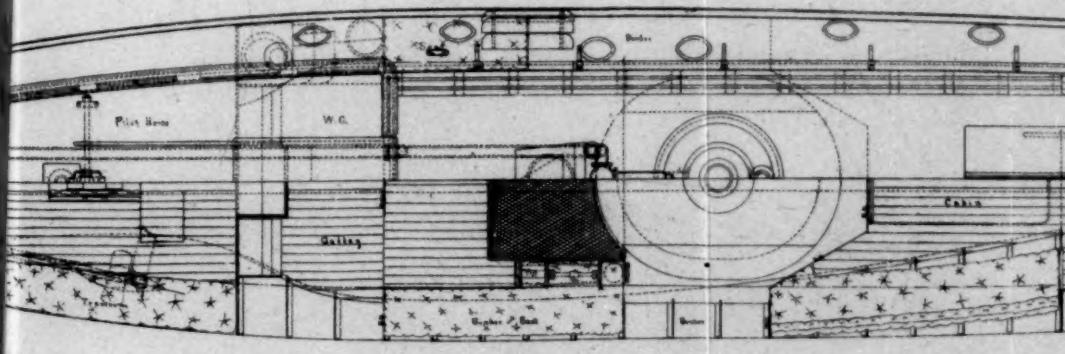
This value is the probable "Atomic Quantity of Electricity." It may be a multiple, but with the same probability with which $C = 12$, and not 6, or 3, is E the quantity of electrical elementary particles. For, even if electricity can be split into smaller parts than E, it is not clear why such a smaller part is never met with in experiment.—*Wied. Annalen, No. 8, 1885; P. Mag., Sept. 1885.*


TWIN SREW IRON STEAM YACHT
for
Dr. Z. Oram of New York,
Designed by William Cowles
ENGINEER & MARINE ARCHT.
111 B'WAY, N.Y. CITY.
NOVEMBER 15th 1884.
Scale, ½ in. = 1 FT.
Dimensions:
After Body Plan
Fore Body Plan
Sheer Plan
Half Breadth Plan
Looking Aft
Looking Forward
Deck Plan



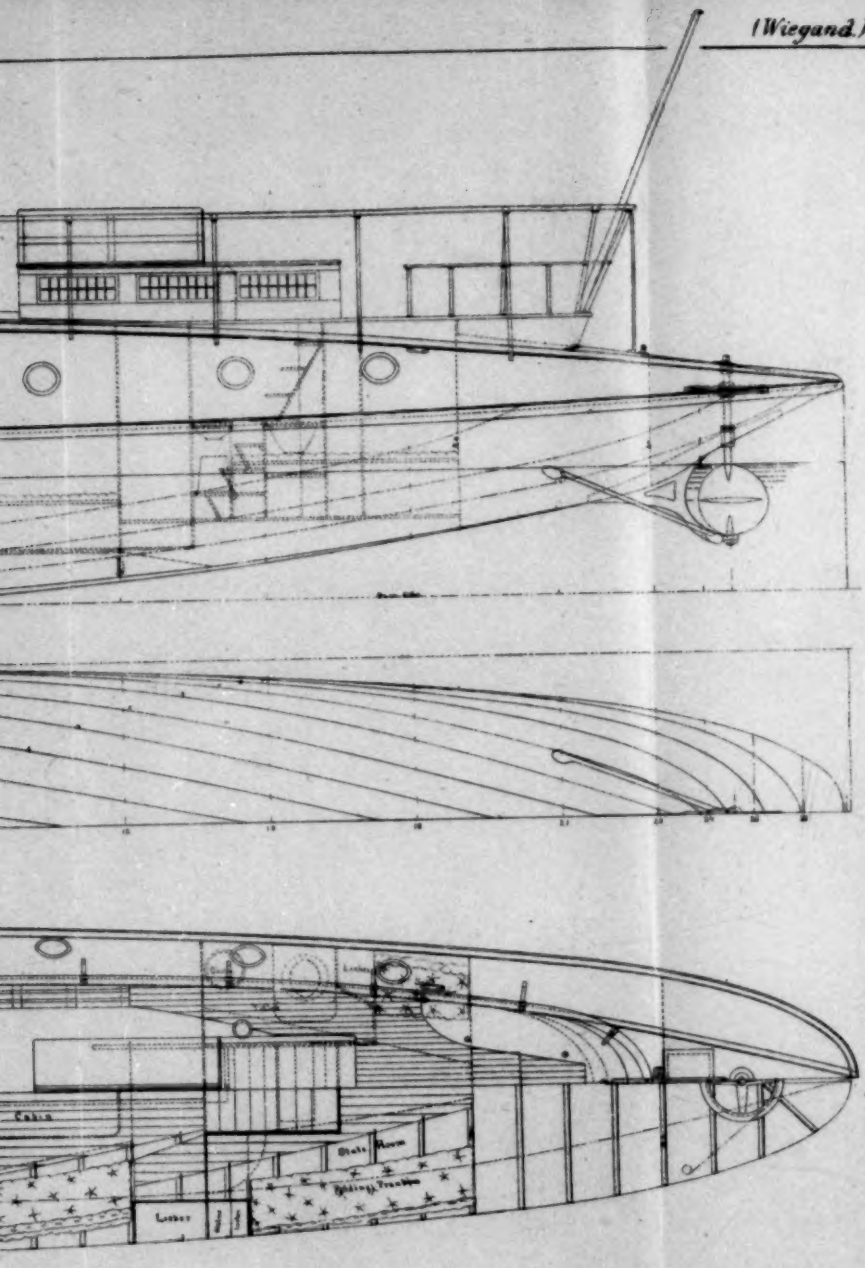
Sheer Plan

Half Breadth Plan



Deck Plan

(Wiegand.)



The ORAM SYSTEM of MARINE PROPULSION.

The Oram System of Marine Propulsion is designed to procure higher speed in steamships with an increased carrying capacity, and to avoid some of the risks of accident incident to propulsion by stern screws.

It consists of an improved form and construction of hull, in which provision is made for a novel location and operation of screw propellers in cavities or recesses on both sides of the vessel.

The propellers are located about one-fifth of the length of the vessel aft of the bow. The cavities in which they are placed are so shaped as to guide the water to and from the propellers with the least possible friction or resistance, and at the same time avoid dangerous protrusion of the propellers beyond the hull.

The propellers are susceptible of rotating independently and are attached directly to the shafts of the engines, having angles of divergence, causing a downward, outward and rearward discharge of the water, attended with a corresponding reaction propelling the vessel.

The recesses are formed of curved surfaces, proportioned and arranged in relation to the propellers so as to guide by the forward curves the incoming water directly to the propellers. The after curves direct the disengaged water in diverging lines, relieving the pressure at the bow and greatly reducing the skin friction of the vessel.

The propellers are susceptible of running at higher speed, require shafts of less diameter than the larger stern screws, and are so short as to be really the engine shafts themselves.

A saving of weight and cost is the immediate result of this form of construction, which not only occurs in the propellers and shafts, but in the engines themselves, which being higher speeded, are of much less weight for the same piston displacement and power.

The long shaft of great weight and the alley to contain and protect it are avoided, and the transverse bulk-heads are unimpaired, thus promoting safety, and the space is available for useful and remunerative storage.

The necessity for longitudinal stiffness in the hull, to sustain

the bearings of the long shaft in proper alignment, does not exist, and the danger from flexure and straining of the hull disabling the shaft and propeller is avoided, so that a degree of elasticity and flexibility of hull construction becomes admissible, which in stern-screw propelled vessels could not be tolerated.

In the event of collision bending or straining the hull the increased security of the machinery against interruption of running is important and valuable, and under some conditions may save the vessel from total loss.

The facility afforded by leaving the hold clear of machinery, to place the cargo or stores of the vessel in compartments separated by bulkheads, is obvious, and for purposes of warfare is important, as permitting constructions not practicable where the shaft alley and shaft divide the lower hold through nearly its entire available length.

The shafts and all of their bearings are constantly under inspection of the engineers on duty, and from their lightness and short length are not so liable to accident as shafts of great weight and length, and couplings being unnecessary accidents thereto are avoided. The danger of racing and breaking, consequent therefrom, is avoided by the location of the propellers.

Economy of fuel in running, of space required for machinery, and in first cost of construction, together with augmented speed and immunity from accident, are claimed for this system.

The accompanying illustration shows a yacht of sixty feet in length, now in course of construction, embodying this invention.

Plans of such a vessel have been submitted to the U. S. Naval Board, having in charge building of the new cruisers authorized by Congress, and are now under consideration by them.

S. L. W.

Philadelphia, April 15, 1886.

DISTRIBUTION OF EARTHY MATTER IN PLANTS.—Berthelot and André have published their second, third and fourth memoirs on the general march of vegetation in an annual plant. They find that the mineral matters which become insoluble by incineration have a marked tendency to accumulate in the leaves. In plants with a languishing vegetation, however, they sometimes seem to be arrested in the roots, probably in consequence of the insufficient action of the agents which render them soluble, and enable them thus to reach the leaves through the circulation of the sap.—*Ann. de Chim. et de Phys., Aug., 1885.*

SOUTH STREET BRIDGE.—NEW WEST APPROACH.

By G. Whitefield Chance.

The South Street Bridge, crossing the Schuylkill at Philadelphia, is one of the few bridges in this country erected on large cylindrical iron columns, filled with masonry, as piers. Two other bridges with such piers are well known, viz., that across the Harlem River, and the great Atchafalaya Bridge, on the Texas and Pacific Railway, built by Major Anderson.

It is very doubtful whether this method of obtaining supports for trusses possesses any material advantage over the ordinary caisson and masonry pier method; while for arch abutments, where the horizontal thrust comes into play so powerfully, this style of pier, as at present built, is of course not advisable.

The bridge, which it is our purpose to describe, was built between the years 1872 and 1876 under a commission authorized by an act of the State Legislature. It was erected in opposition to the wishes of City Councils, who filed a protest against the action of the Legislature.

Moses A. Dropsie and ex-Judge Findlay were the successive Presidents of the Bridge Commission. A contract to build the structure was made by the Commission with John W. Murphy, C. E., on March 30, 1870, for $770,000. The total cost amounted to $865,000.

Mr. Murphy died about a year before the completion of the bridge, and it was finished by his executors.*

The somewhat great length of time consumed in the erection of the structure was mainly due to legal delays.

GENERAL DESCRIPTION.

The bridge and approaches have a total length of 1,934 feet 7 inches. The bridge proper consists of two 195 feet 8 inch fixed spans, of the Murphy-Whipple form of truss, and a draw 198 feet 2 inches long, of the same type between.

The depth of all the trusses is twenty-seven feet.

* See biographical notice of John W. Murphy, C. E., by W. Barnet Le Van, read before the Franklin Institute, Philadelphia, October 21, 1874.

The shore spans have a breadth of thirty-six feet between trusses, while the total width between railings on the outside of footways is fifty-five feet.

The columns of the trusses are of the Reeves patent.

The shore ends of the fixed spans each rest on stone abutments, while the river ends each rest on two eight feet in diameter iron columns. These columns are braced laterally, and the space between them filled in with loose stone and masonry.

The columns are of cast iron, and are in sections 10 feet in length, 1¾ inches thick, bolted together by inside flanges at each end, 2¾ inches wide and 1¾ inches thick. The flanges are pierced with holes five inches apart from centre to centre, which are filled with one and one-fourth inch bolts. The sections resting on rock are cut-off square and secured to the rock by means of iron brackets and fox wedges. Each ten feet section weighs 14,600 pounds. The inside of the columns is filled with rubble masonry.*

PIVOT PIER.

The pivot pier is formed of eight columns, four feet in diameter spaced equally around a circle of twenty-eight feet diameter, and tangent thereto. These columns carry the track on which the wheels of the turn-table run. The pivot rests on a six foot in diameter column, and has a Sellers' Patent Centre.

The six foot column weighs 10,800 pounds per section of ten feet, and the four foot column, which is one and one-fourth inches thick, 6,800 pounds per ten foot section.

The columns average about sixty feet in length. They are all braced vertically and horizontally by I-beams and tie rods, and below low water they are surrounded by a cribwork filled with rip rap. The load on the pivot pier from the draw span when open, is 400 tons.

The clear water way on each side of the pivot pier is seventy-seven feet.

Piers of masonry are placed up and down stream eighty-eight feet from the centre of pivot pier for supporting the draw when open, and also to serve as guard piers.

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*For full description of method of sinking cylinders, see article by D. McN. Stauffer JOUR. FRANKLIN INSTITUTE, Vol, 64, p. 320.

EAST APPROACH.

Commencing at the river and going east is the abutment pier, 40 feet 10 inches in length, built on a hard gravelly soil; it is of granite with pilasters and Doric capitals.

Then come three conoidal brick arches of original design built on a line which curves to the left. The curve has a radius of 160 feet, and the central angle subtending the arc between the tangents at beginning and end of curve is 33° 25′, the length of curve being 114 feet 6 inches. The facing of these arches is of brown sandstone with granite trimmings, the piers being granite.*

After the arches, there is 363 feet 6 inches of broken range ashlar retaining wall of brown sandstone with granite trimmings, the approach descending at a steep grade to the level of the street. The total length of the approach is 518 feet 10 inches.

OLD WEST APPROACH.

Commencing at the river and going west was an abutment 40 feet 10 inches in length, similar to the eastern abutment, placed on 517 piles driven butt down, and connected by a timber grillage. Then came nine segmental brick arches, each of 43 feet 6 inches span, and 14 feet rise, 2 feet deep, placed on granite piers 5 feet 6 inches wide. These arches covered a distance of 391 feet 3 inches. After the line of arches was an abutment pier of granite 62 feet 4 inches long. Between this pier and the granite retaining wall 87 feet 4 inches long ending the approach, were three Pratt trusses, 244 feet 9 inches total length, resting on eight wrought iron columns. These trusses carried the approach across the Philadelphia, Wilmington and Baltimore and the tracks of the West Chester Railroad.

In the new west approach, the above enumerated features remain intact with the exception of the line of arches with their piers and the railroad abutment pier.

The total length of the west approach is 826 feet 6 inches.

PROFILE.

The general character of the soil along a profile section through the centre line of the bridge, is as follows: At the surface, tough

* For full description of conoidal arches, see "Proceedings of Engineers' Club of Philadelphia," Vol. I, p. 95.

mud extending to considerable depth in some places, then coarse gravel underlying, then very coarse gravel, and finally an undulating micaceous gneiss. Lying directly on the rock in the river bed, are considerable quantities of drift wood, evincing great age and a long occupation by the river of its present position.

The rock which at the eastern river abutment is five feet below the bed of the river, or seven feet below low water, shelves off to eighteen feet at the draw pier, or thirty-four feet below low water, while at the western column pier it is thirty feet below the bottom or thirty-eight feet below low water.

The iron columns were all sunk to rock by means of the plenum pneumatic process.

CRACKS IN COLUMNS.

Some of the iron columns of the pivot pier have cracked badly.

Some of these cracks occurred shortly after the bridge was erected.

The columns were cast from Government cannon, originally made from cold blast charcoal pig.

The cracks commence at the joints and extend both ways in some instances, generally vertically, as far as six feet from the joints. They vary in width from zero to two and one-half inches.

It is most likely that the cracks were caused by water freezing inside of the columns. The force exerted by water in freezing, or in its effort to freeze when confined, is probably as great as 30,000 pounds per square inch. (Trautwine.)

From the fundamental formula in hydromechanics,

$$e = \frac{r\,p}{T}$$

in which,

e = the thickness of a pipe to resist longitudinal rupture,
p = the pressure on unit area,
T = ultimate tensile strength of material of pipe,
r = internal radius of pipe,

we may calculate how thick a pipe or column would be required to sustain a pressure of 30,000 pounds per square inch. Taking the ultimate tensile strength of the metal of Government cannon at 38,000 pounds per square inch, we find that the column should have a thickness of about eighteen inches, while it has only one

and one-fourth inches. This theoretical thickness is probably too great, owing to the fact that expansion could take place and so relieve the pressure.

Water could have entered the columns in several ways, either

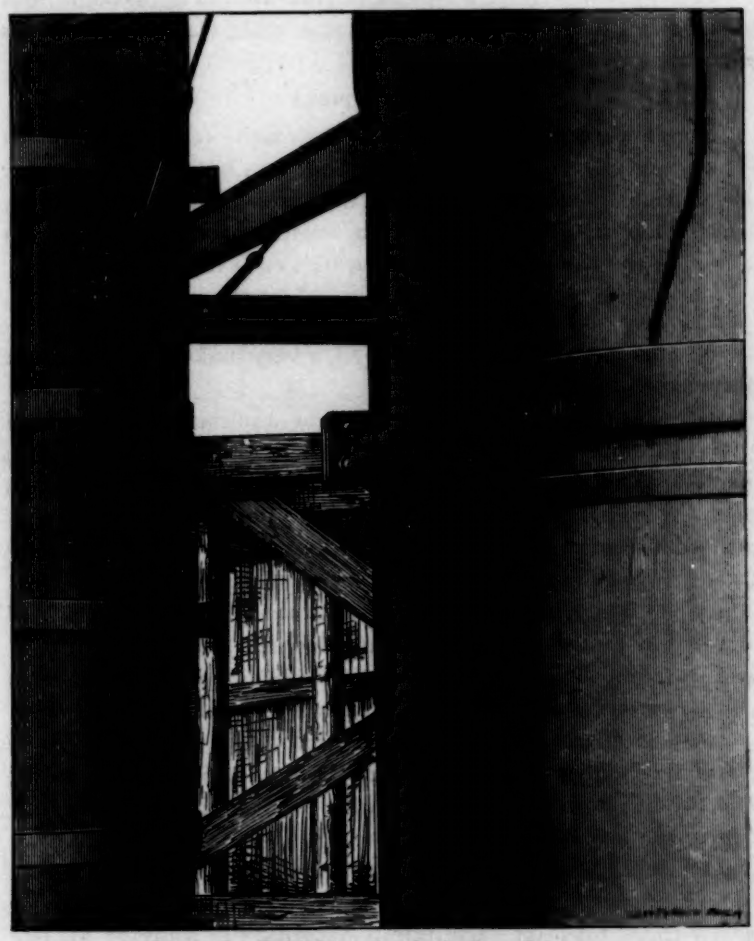


PLATE I. Exhibiting Cracks in Iron Columns.

at the section joints, from the top, or through holes in the columns, four of which are shown under the I-beam in the column on the left of *Plate I*, which is an engraving taken from a photograph by Prof. L. M. Haupt. A somewhat significant fact is that out of nine

large cracks in the pivot pier columns, six are below the level of high stages of the river.

The verticality of the cracks shows a horizontal force of some kind acting radially from within.

The cracked sections have been bound around by wrought iron bands ¾ of an inch thick by 5⅞ inches wide, clamped with bolts and nuts.

OLD ARCH PIERS.

The arch piers of the western approach were each built on eighty-four piles, not driven to rock, but to a hard gravel stratum. These piles, together with the piles used under the abutment piers, some 1,200 in all, were of Nova Scotia spruce, averaging ten inches on the head, and were driven butt ends down.

FAILURE OF.

On Sunday, February 10, 1878, a very unfortunate accident occurred to the west approach.

Pier No. 7, numbering from the river abutment westward and 385 feet from the river, sank at its northern end some sixteen feet, involving a falling in of its two arches; and since the arch piers were not sufficiently massive to withstand the horizontal thrust of each arch separately without the resisting thrust of the adjoining arch, the whole line of arches fell in ruins.

At pier No. 7, the piles were driven through fifteen feet of mud to a bed of gravel, by a 2,000-pound hammer, and received, as a final test, four blows from the hammer falling thirty feet, when one inch downward motion was the maximum allowance for each blow.

At the large abutment pier, 100 feet west of the failing pier, the gravel bed just mentioned is within a foot of the surface, and it seems to dip from this point uniformly towards the river, where it is forty-five feet below the surface.

D. McN. Stauffer, Assistant Engineer of the bridge, says: "The estimated load on the piles under the pier was 2,000 tons, or twenty-three tons on each pile. According to Rankine, for pile work under similar circumstances, each pile was good for eight tons frictional value, leaving fifteen tons on the toe of the pile. The tremor produced in the piles by travel on the bridge would gradually loosen them sufficiently to allow percolation of water down their sides, and finally throw the whole weight on the toe of

Elevation of Girder Approach
—to—
SOUTH STREET BRIDGE.

Scale.

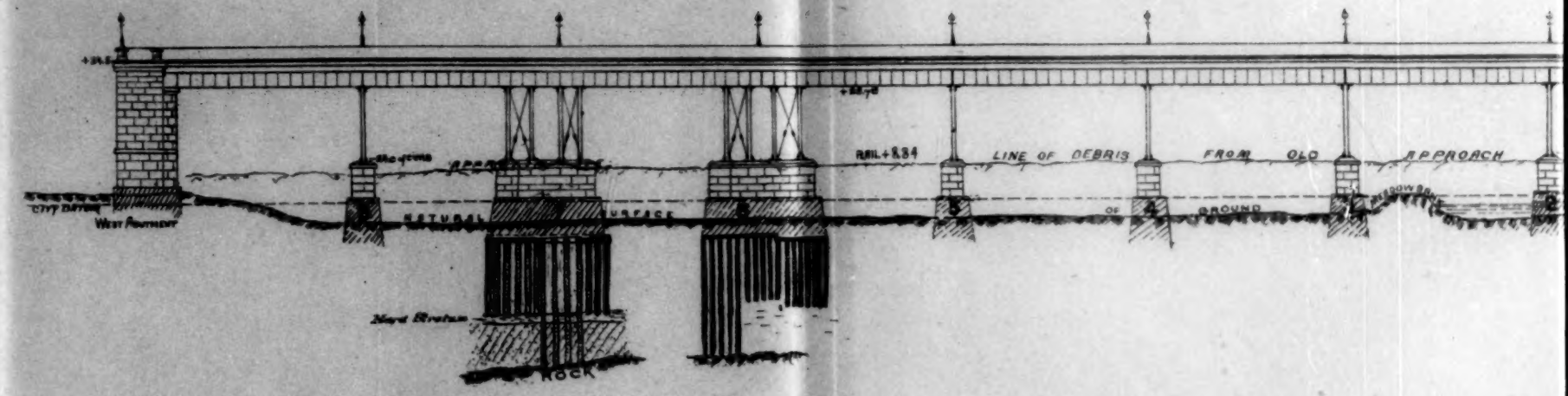


PLATE II.

Elevation of Girder Approach
—to—
SOUTH STREET BRIDGE.

Scale.

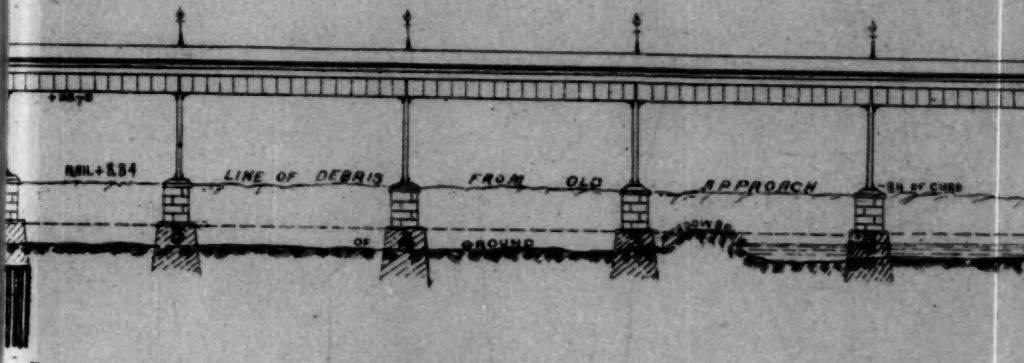
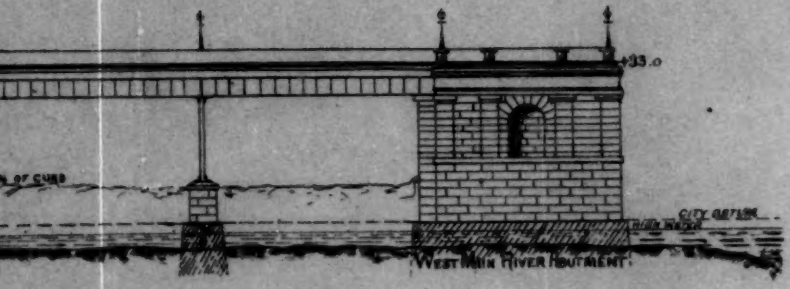


PLATE II.

(*Chance.*)



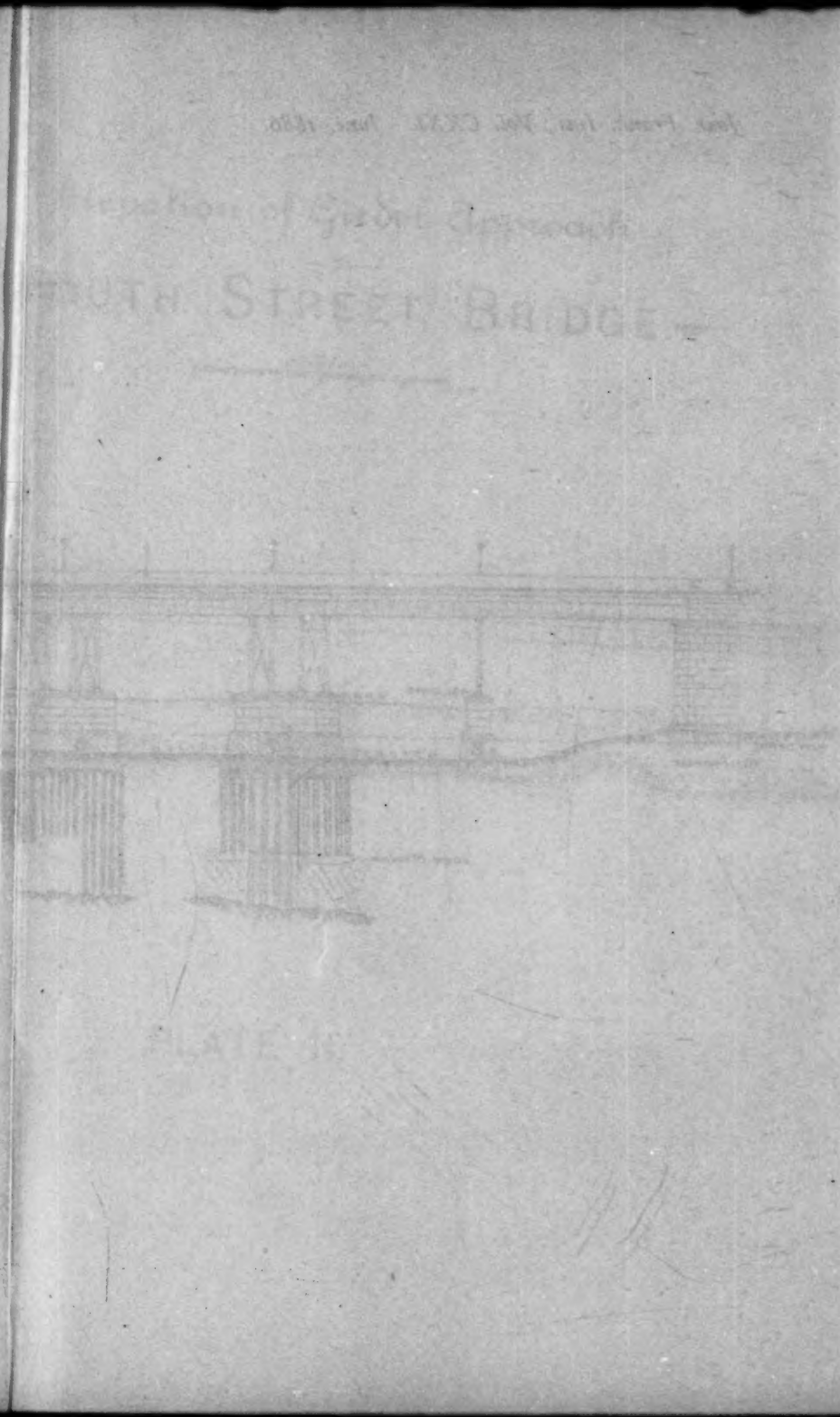


Plate Girder Approach to SOUTH STREET BRIDGE.

PLA

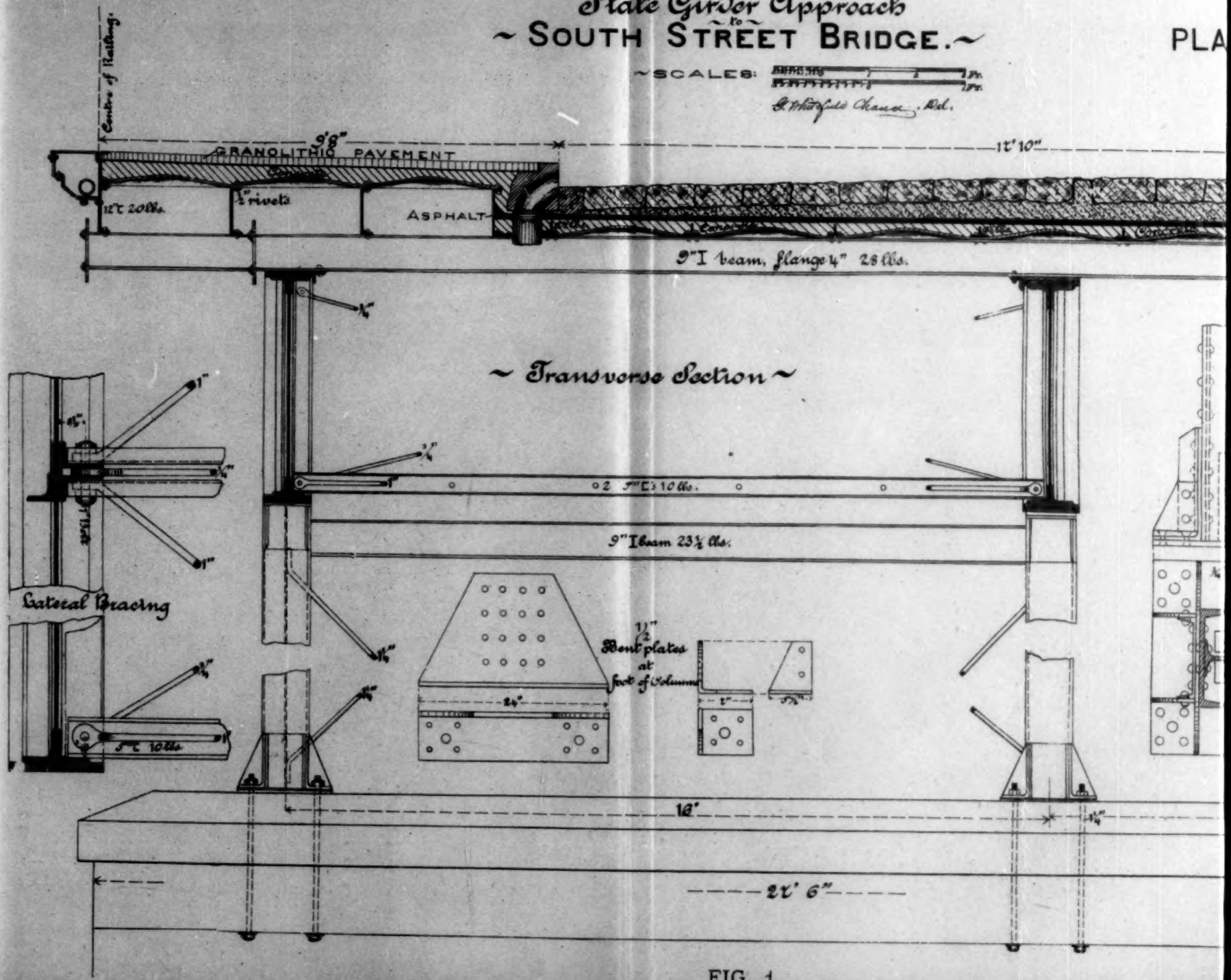


FIG. 1.

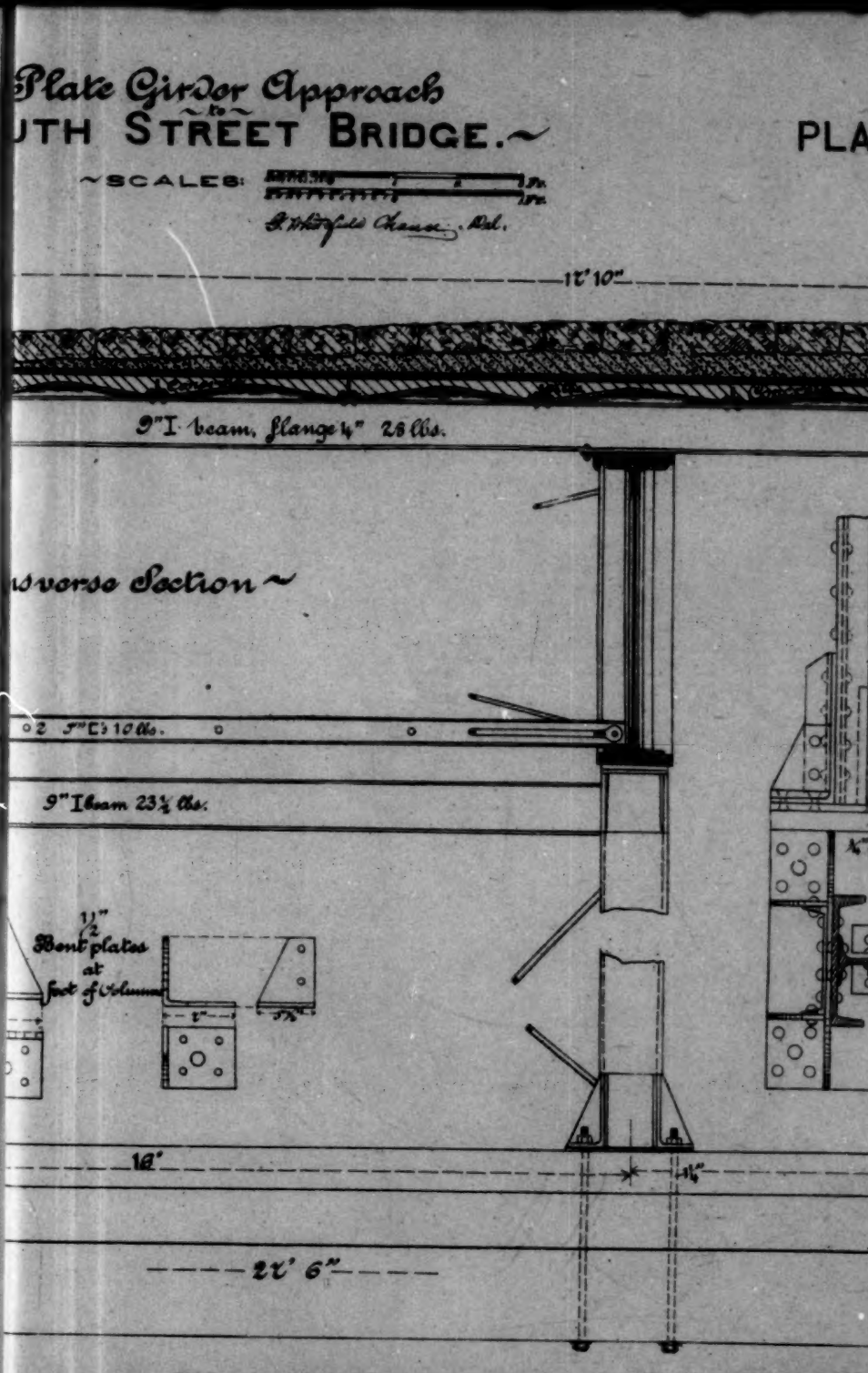
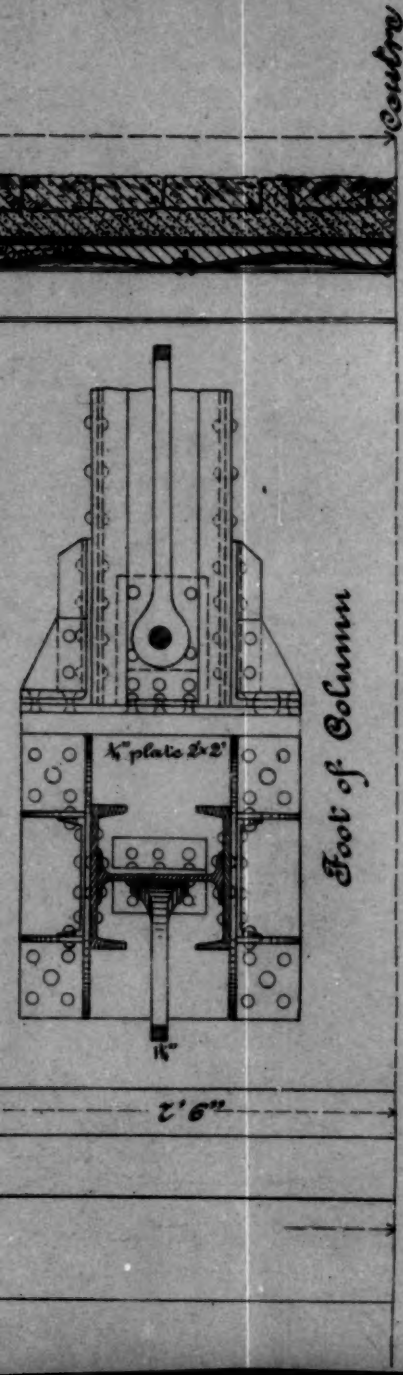
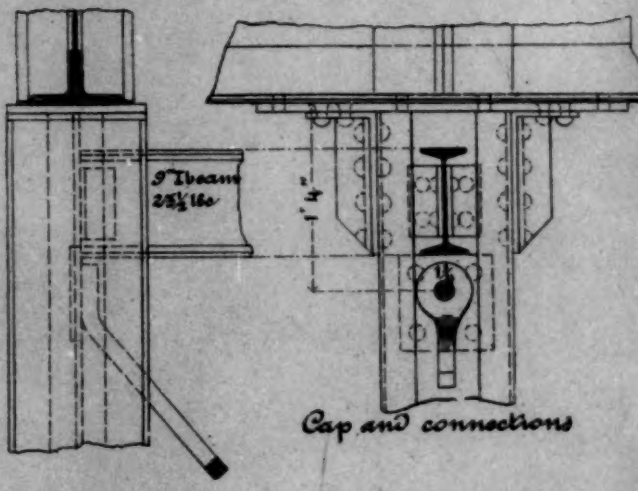


Plate Girder Approach
to
UTH STREET BRIDGE.~
PLA
~SCALES:
P. Whitfield Haus, Del.
17' 10"
9" I beam, flange 1/4" 28 lbs.
isverse Section ~
2 3" ['s 10 lbs.
9" I beam 23 1/2 lbs.
1 1/2"
Bent plates
at
foot of Column
1"
3/8
16'
1 1/4"
22' 6"

FIG. 1.

PLATE III





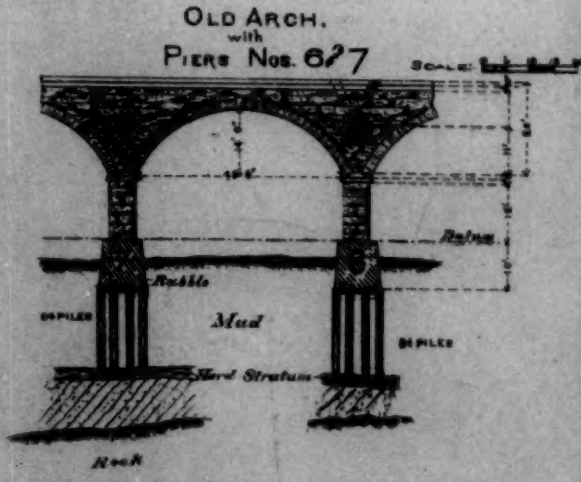
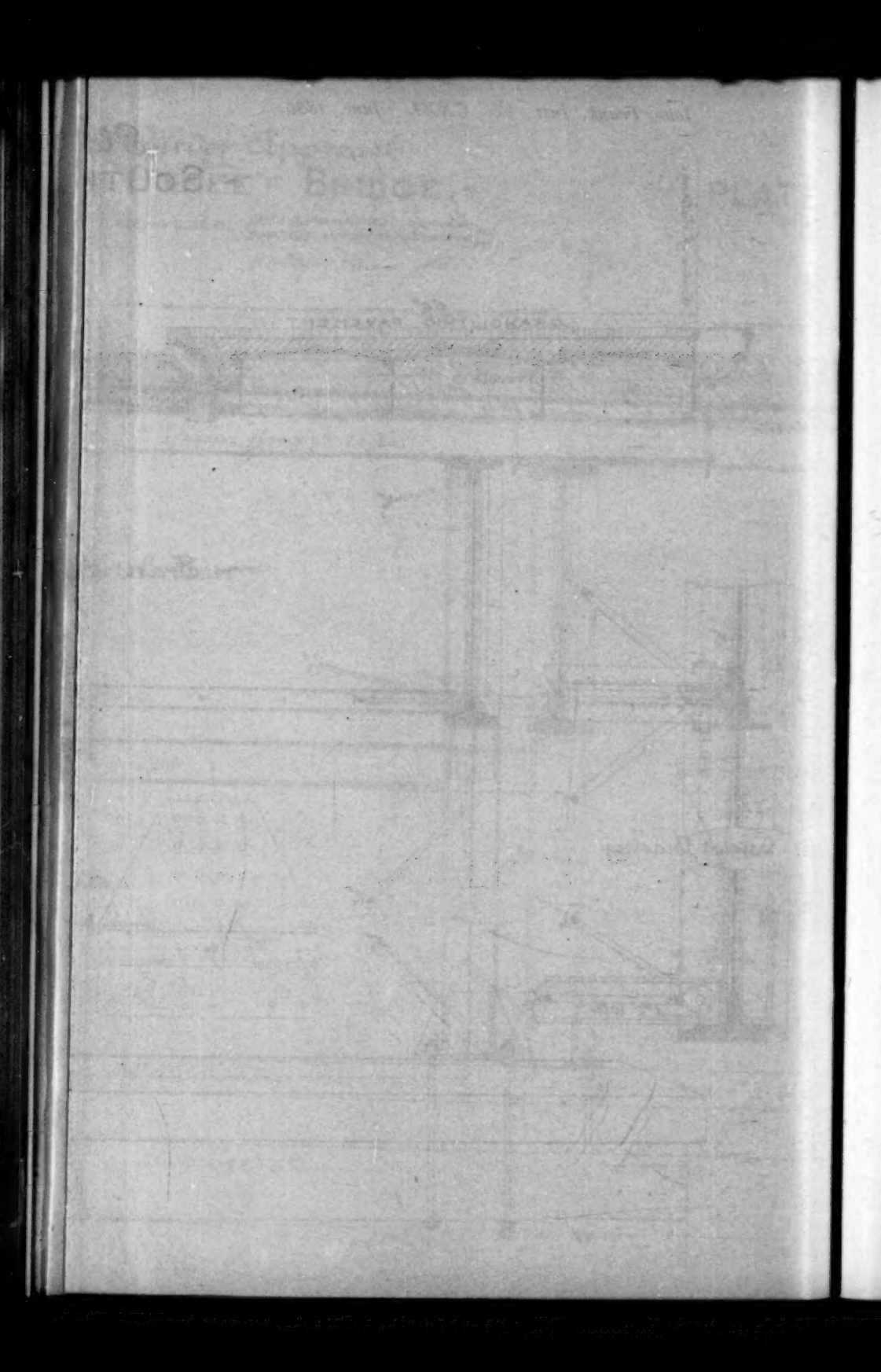


FIG. 2.



the pile. If the bed of gravel was merely a thin bed, or pocket, as the water softened it the additional weight, before sustained by friction, probably allowed a bunch of piles at the north end of the pier to settle down into the soft material beneath the gravel." He also says:

A small expenditure to put in tie rods between the two adjacent piers to No. 7 would have saved seven out of the nine arches. Borings made twenty feet from each end of the failing pier showed, at the south end gneiss 33 feet 8 inches below the surface, overlaid by from eighteen inches to twenty inches of gravel and then a considerable thickness of a hard, very compact, yellowish clay.

At the north end, rock was found thirty-seven feet below the surface, above which was 3 feet 8 inches of black mud and above this seven feet of gravel mixed with clay.

In excavating for the new foundations of pier No. 7, the largest pile found was fifteen feet in length. Some of the piles of the east row were found to have their ground ends higher than their heads, the piles pointing in a northeasterly direction (the pier runs nearly North and South), making an angle of about 30° with the horizon. The piles of the row next to the east row were found to have been broken off two feet below their heads, the remainder being held in place by the concrete surrounding.

The diagram of old arch with piers Nos. 6 and 7 (*Plate III. Fig. 2.*) will render it easy to make calculations for the total load on each pile.

The cut stone masonry of the face of the arch, including the arch rings, spandrel walls and coping, extended in on an average three feet. The arch was of brick while the haunching, as shown by the dotted lines, was of rubble. Above the haunching was gravel filling, on top of which was a granite pavement four inches deep. The piers were of granite, while the foundations were of rubble. A timber grillage, some two feet thick, covered the piles. The barrel of the arch was fifty-five feet long.

From Trautwine's formula for safe load for piles, we have:

$$L = \frac{F \times \cdot 023 \sqrt[3]{h} \times w}{s + 1}$$

in which,

L = safe load in tons;
h = fall of ram in feet;
w = weight of ram in pounds;
s = last sinking of pile in inches;
F = factor of safety.

Using the quantities before given, $h = 30$ feet : $w = 2{,}000$ pounds, and $s = 1$ inch, we find, taking one-third as the factor of safety, which is large considering the conditions there present, the safe load would equal 23·8 tons per pile.

NEW GIRDER APPROACH.

A temporary wooden structure was used for travel during the construction of the present handsome girder approach, completed December, 1885.

The new girder approach, which takes the place of the old arches, consists of 446 feet of wrought-iron plate girder bridge work, 57 feet in width, with a roadway 35 feet 8 inches, and two footways, each 9 feet 8 inches in width.

It is divided into nine spans, supported by wrought iron columns on masonry piers. Six of the piers are built on the old foundations, while the seventh or failing pier has an entirely new one. (*See Plate II.*)

The railroad abutment pier adjoining the arches was reduced in length to fifteen feet.

The foundations of pier No. 6 were strengthened at its south end and its plan was changed to make its transverse centre line oblique to the centre line of bridge, in order to accommodate certain railway tracks; this involved in all a driving of forty-six piles.

Pier No. 7 was built over again, and its plan was also changed so as to make it parallel with pier No. 6.

Eighty-eight new piles were used in its foundations, from 22 to 30 feet long and about 12 inches in diameter on the head, some of which were driven to rock by a 2,200 pound hammer. These piles were of oak, straight growth and barked, and were originally about forty feet long when driven, but were cut off to –9 feet below datum.*

* City datum, $8\frac{52}{100}$ feet above mean low water, Delaware R.

The spaces between these piles were filled for three feet below their heads with concrete. They were capped transversely to the line of bridge with 12 x 12 inches hemlock timbers, fastened to the piles with locust tree nails 1¼ x 24 inches, and a platform 10 x 60 feet of 10 x 12 inches hemlock timbers at right angles to the caps, laid close, was then secured to the cap pieces at crossings by 1¼ x 18 inches locust tree nails.

Rubble masonry 9 x 59 feet was then laid on the platform and carried up with a batter of about one foot on two-thirds inch all around to within –33·5 feet of curb on roadway. Five courses of pitched ashlar, each about two feet deep, were then laid on and covered with a coping eighteen inches thick. Each pier is 5 feet 6 inches wide, and has sixteen bolts and eight plates built into it for anchoring the iron columns; there are two plates and four bolts to each column.

PLATE GIRDER SPANS.

The plate girder spans are each about forty-nine feet in length, and are formed of four lines of girders each about four and three-fourths feet deep, resting on four wrought iron columns at each pier. Each column is formed of two twelve-inch channel bars, weighing thirty pounds per lineal foot, connected on the flange side of their webs by a ten and one-half inch **I**-beam, thirty pounds per lineal foot. The columns are bolted to the piers by four one and one-fourth inch wrought iron bolts, extending through the coping and two additional courses of stone, and connected by two tie plates, each 4 x ¼ x 24 inches.

The inside lines of columns are placed sixteen feet transversely on the piers from the outside lines, while the inside lines are fifteen feet apart. Each column of the outside lines is braced with each column of the inside lines, on each of the piers, by a nine-inch **I**-beam, twenty-three and one-half pounds per lineal foot, placed at the top of columns, and the panel formed by the two columns, **I**-beam and pier, is tied with two one and one-fourth inch square wrought iron rods, with turnbuckles, placed diagonally

The outside and inside lines of girders are connected over each pier and half way between spans by vertical panels of bracing, each consisting of two five-inch, ten-pound channel bars, placed at the bottom of the webs of the girders, and two three-fourths inch round wrought iron rods, placed diagonally, and running

from the top of one girder to the bottom of the other, with connections, as shown in *Plate III. Fig. I.* Each horizontal panel formed by the two girders and the two five-inch channels, just mentioned, dividing the span into two equal parts, together with the two five-inch channels over each pier, is tied by two diagonal round wrought iron rods, each one inch in diameter. All transverse rod bracing is tied at intersections by copper wire.

There is no bracing of any kind connecting the inside lines of girders.

At every second pier a distance of about one-half inch is allowed between the ends of girders for expansion, and slotted holes are made to give the bolts proper play.

FLOOR FORMATION.

On the girders in each span are placed, transversely, eighteen wrought iron nine-inch I-beams, flange four inches, twenty-eight pounds per lineal foot. The middle and two end beams in each span are riveted to the girders. Riveted to the I-beams mentioned, longitudinally, on each side of the bridge, to form the footways, are four twelve-inch channels, twenty pounds per lineal foot, placed 2 feet 9 inches apart; on the top of each outside line of channels, is riveted an eight inch, twelve and one-half pound channel to retain the footway surfacing and concrete. To this channel and the one below, the cornice, formed of channels and angles weighing about sixty pounds per lineal foot, is riveted. On the ends of the floor I-beams, ornamented castings are affixed.

Upon the channels and I-beams of the floor formation, are placed buckled plates, riveted thereto on two edges with one-half-inch rivets. These plates are of wrought iron ¼ inch thick and 3 feet square with 2½ inch rise in the middle.

Upon the unsupported edges of the plates are placed tee bars, six and one-half pounds per lineal foot on roadway, and five and one-half pounds on footways. The flanges of these tees are riveted to the edges of adjoining plates.

At piers Nos. 1, 2, 4, 6 and 8, expansion joints are made in the floorway formation. Each is formed of two angles riveted to the upper flanges of two adjacent I-beams, and a plate, ⅜ inch thick by 15 inches wide, to which 2 x 2-inch angles are riveted, placed upon the upturned edges thereof, with the angles turned down.

SURFACING.

Upon the buckled plates, up to one inch over their apices, is placed a layer of concrete, formed of two parts of gravel and one of German Portland cement, upon this is a layer one inch thick of Neuchatel asphalt composition, ironed to make water tight joint —arranged as shown in *Plate III. Fig. I.* A layer of concrete, four and one-half inches deep in the middle of roadway, covers the asphalt. The concrete is of the usual formula, viz., stone, four parts; sand, two parts; cement, one part.

On this concrete, is a layer of the first mentioned concrete two inches deep (not shown in *Plate III*), in which the granite blocks of the surface covering are laid. These blocks are about 6 x 9 inches in breadth and length and 3 inches deep. No joints over three-fourths inch wide were allowed by the specifications. As soon as the top layer of cement mortar had set, the joints of the pavements were grouted with Portland cement and sand half and half. No travel was allowed for two weeks after grouting. No stones over one and one-half inches in their largest dimension were allowed in the concrete of the superstructure.

The footways are formed of a layer of concrete about two inches deep over the apices of the buckled plates, on which is laid two inches deep, "Stuart's Granolithic Pavement," forming a most excellent surface.

Over piers Nos. 1, 3, 5 and 7 are projecting lookout bays in the footway formation. Each span of superstructure weighs 400 tons.

SPECIFICATIONS.

The specifications stipulated in regard to the work as follows:

* * * * * * * * * *

(49.) Portland cement must weigh 112 pounds per struck U. S. bushel of 2150·42 cubic inches. It must be ground so fine that no residue remains after passing through a wire cloth sieve 3,600 meshes per square inch.

After being made into a briquette and remaining seven days in water it must show a tensile strength of not less than 200 pounds per square inch.

Other cement must not leave more than fifteen per cent. residue on wire sieve (3,600 meshes per square inch), and exhibit a

tensile strength of seventy-two pounds per square inch after remaining in water seven days.

* * * * * * * * * *

(51.) All mortar is to consist of cement and sand in the proportions of one part cement, two parts sand.

* * * * * * * * * *

(54.) All wrought iron shall be uniform in quality, tough, fibrous, free from cracks and flaws and with perfect edges.

All wrought iron in tension shall be capable of withstanding a tensile stress of 46,000 pounds per square inch of full section of specimen, with fifteen per cent. (web plates five per cent.) extension in eight inches, and of supporting fifty per cent. of that weight without permanent set.

All lateral and transverse tie rods shall have adjustable sleeve nuts and connections equal to the strength of the rods.

(55.) In all rivet work, the holes shall be accurately spaced and punched truly opposite. The rivets shall be of the best quality of rivet iron; shall completely fill the holes and have hemispherical heads.

* * * * * * * * * *

(58.) All new iron work shall be cleaned and receive one coat of red oxide of iron ground in pure boiled linseed oil before leaving the shops, and after erection shall receive two coats of the same tinted dark brown or black, as directed; the pigments also being ground in oil.

The contract price paid by the city for the work was $81,920. The work included some repairs to the abutments and some other minor details necessary to place the approach in complete order for travel.

Work was commenced about the middle of June, 1884, and the approach was thrown open in the early part of December, 1885.

My acknowledgments are due to Mr. Samuel L. Smedley, Chief Engineer and Surveyor of Philadelphia; Mr. J. Milton Titlow, Principal Assistant Engineer, and Mr. Lindley M. Winston, Inspector, who kindly furnished me with many of the data embraced in this paper.

EXPERIMENTS ON THE TRANSMISSION OF POWER BY GEARING.—MADE BY MESSRS. WILLIAM SELLERS & CO.*

By Wilfred Lewis, Philadelphia, Pa.

[*A Paper read before the American Society of Mechanical Engineers, at the Boston Meeting, 1885.*]

The idea of determining, by experiment, the relative values of different forms of gearing, as measured by the economy with which they transmitted power, originated with Mr. J. Sellers Bancroft, in the spring of 1883, and soon after the following investigation was begun at his instance and carried out under his direction. The question implied was one of continual recurrence, upon which diverse opinions were found to exist, and these were so frequently based upon general impressions or loose observations, in which some important fact or condition was wanting, that the necessity was felt for more definite and reliable information as a guide in the construction of high-class machinery. To obtain this in such a manner as to leave no room for doubt, it was proposed that the gearing itself should be made to bear testimony, and the writer accordingly undertook to prepare the design for an apparatus which should thoroughly test the efficiency of the more common forms of gearing, namely, worm, spiral and spur.

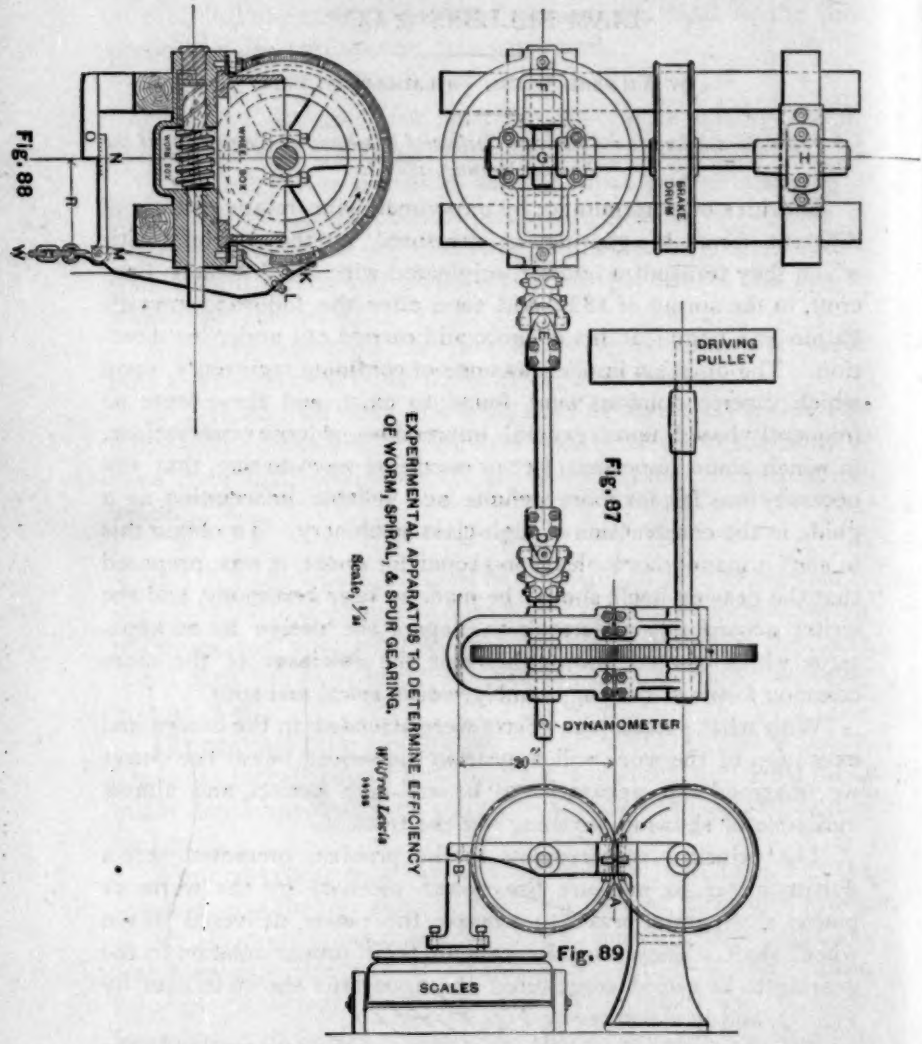
With what success our efforts were attended in the design and execution of the work will appear in the sequel, but at the outset we imagined our apparatus to be well-nigh perfect, and almost incapable of showing anything but the truth.

The principal requirements in the problem presented were a dynamometer to measure the power received by the worm or pinion shaft, and a brake to measure the power delivered to the wheel shaft. These two elements, in their proper relation to the gearing to be tested, constituted the apparatus shown in plan by *Fig. 87*, and in elevation by *Figs. 88* and *89*.

The chief difficulty in the perfection of the scheme lay in the construction of a dynamometer which should not have the errors and defects common to such machines.

* From Advance Sheets, Vol. VII., Transactions A. S. M. E.

We required, in the first place, that it should be readable at any time, with as much delicacy and precision as an ordinary pair of scales; and, in the second place, that its own resistance should,



if possible, be eliminated from its readings, without the necessity of making guess-work corrections.

An experimental dynamometer, similar to the one represented,

was built at the suggestion of Mr. Bancroft, and tested with encouraging results; but, instead of the universal joints, he used simply a straight shaft and depended upon its flexibility, and instead of the frame hinged at *A*, two separate frames were used and the shafts held at a fixed distance apart by means of links.

From a practical point of view, this might have been considered a success, but, theoretically, it was subject to an error arising from the friction of the driven shaft in its bearings. When properly lubricated, this friction would be small, but from carelessness in oiling, it might well be worth considering, and the arrangement shown in *Fig. 87* was designed as an improvement. Here it will be seen that the frame which carries the driven wheel is supported by a flexible joint at the line of contact *A*, and again at *B*, by portable platform scales.

From this construction, it follows that no matter how great the friction in the journals of the shaft *C D* may be, there will be no pressure at the point *B*, except what results from torsion in the shaft *D E*.*

To demonstrate this point, let us imagine the driven wheel firmly clamped to its frame, so that for all intents and purposes it may be considered as part of the lever *A B*. Now, it is evident that a downward force acting on this lever at any point to the right or left of *A*, will produce at *B* a reduction or increase of pressure proportional to the distance of the force applied from *A*. And it is likewise apparent, when a force acts directly in line with *A*, that no portion of it can be communicated to *B*. When this principle is clearly understood, it can readily be seen that journal friction is nothing more than a partial clamping of the wheel to its frame *P*. A thorough test of the accuracy of this construction was made by attaching to the shaft *C* a brake in connection with the frame *A B*. When the wheels were set in motion, this brake represented a magnified case of journal friction, and no matter how tightly it was drawn, the reading of the scales remained unaltered.

As a still further refinement of this dynamometer, the attempt was made to avoid the use of gear teeth altogether and drive by the contact of flat-faced wheels, but the great pressure required to

* This principle is the same which was used by Mr. Tatham in his belt dynamometer.—[W. L.]

drive caused the journals to heat rapidly, and the idea was soon abandoned. Otherwise the method was satisfactory, and the truth of the principle just demonstrated was repeatedly shown by the fact that the reading of the scales could not be altered by any change in the clamping pressure on these discs.

To measure the power transmitted, it was important to know the effective length of the lever arm *A B*, and this was carefully tested in the following manner and found to agree with actual measurement.

The universal joints were disconnected, the driving pulley was blocked, and a weighted lever attached to the shaft *C*; the scales were then balanced, and the weight was moved out upon the lever until the reading of the scales was increased by an amount equal to the weight moved. The distance through which the weight had to move was then taken to be the effective length of the lever arm of the dynamometer. The brake used to measure the power transmitted was designed to maintain an approximately constant load of any desired amount. It consisted of a drum or pulley encircled by a Napier brake of rather novel construction, in which the load itself was made to adjust automatically the tension on the strap.

By this means, there were no extraneous forces involved and no corrections to be made, and the error usually arising from the use of an auxiliary tightening lever was avoided.

By reference to the drawing, it will be seen that, for any given load suspended at the point *M*, the tension on the strap at *L* depends upon the horizontal distance of *M* from *K*, and that the moment of resistance is measured by the horizontal distance *R* of the point *M* from the centre of the drum.

As a precaution against excessive heating during the progress of experiments, this brake drum was provided with deep internal flanges forming a trough through which water could be made to pass by means of a siphon.

The gearing to be tested is represented on the drawing as a worm and wheel, but the housings which carry their journals are adjustable, so that the brake shaft can be set obliquely to the worm shaft for spiral gearing, or parallel to it for spur gearing.

The worm box was made very large, to insure an abundance of oil and dissipate as fast as possible the heat of friction. It also

acted as a reservoir for the lubrication of the journal and step-bearings, and as a bath in which a thermometer could be kept to note the temperature from time to time.

Power was received by a belt to the driving pulley from an independent engine, the speed of which was under the control of the experimenter. In preparing for experiments, the shafts *C D* and *E F* were set in line, or nearly so, and the edge of the block *O* adjusted to be directly under the centre of the drum shaft. Then, with a known weight attached to the chain from *M*, the worm box filled with lard oil, and the apparatus set in motion, the experiments were ready to begin.

By means of the nut *L*, the position of the weight was adjusted until the graduated bar *M N*, resting on *O* and *M*, was brought to a horizontal position. Then the revolutions per minute were counted, the temperature noted, and the reading of the dynamometer observed, together with the distance *R* on the bar *M N*.

In commencing to make the experiments, several unexpected difficulties were encountered, which for a time vitiated the results to an unknown extent. The experiments thus affected, however, were afterward repeated so as to exclude any errors which might have crept in unobserved.

At moderate speeds everything ran smoothly, but as the speed was increased, the step bearing for the worm shaft began to give trouble. This bearing was formed by the contact of two hardened steel discs carefully ground, and although well lubricated by a circulation of oil, the danger of cutting at high speed was constantly to be apprehended.

The delay caused in this way was very annoying, and after a number of futile efforts to lengthen the life of the bearing, the experiment was finally made of introducing a loose washer of hard brass between the hardened steel faces.

This remedy proved to be effectual, and no further evidence of cutting at this point was detected for any combination of speed and pressure.

The brake strap was also at first another source of annoyance. It was made as shown by the drawing of a flat iron strap lined with hard wood blocks on end grain, and for a time it worked well, although not quite as steadily and smoothly as could be desired.

There appeared to be considerable variation in the friction of these blocks against the drum, and the weight suspended was in consequence thrown into oscillations, which had to be checked before the distance R could be properly measured.

To check these oscillations it was simply necessary to press the bar $M N$ against its bearing, until they were absorbed by the friction thereby produced, and so long as a slight amount of oscillation remained, it was considered as evidence that this friction did not effect the mean radius to be determined. Still there were variations in the amplitude of the vibrations which could not be altogether controlled in this way, and for the purpose of improving its action the brake strap was lubricated with a mixture of tallow and oil, which at once produced the most alarming results. The load was thrown into the most violent agitation, and the apparatus shaken in a manner which threatened to break it down. For this strange phenomenon no satisfactory explanation could at first be given. The apparatus was strengthened and braced in various ways, sometimes with an improvement in its general working but oftener without avail. The vibrations were at times very violent, and again entirely absent under apparently the same conditions. Sometimes they increased in violence with the speed, and sometimes they diminished, and finally disappeared.

At or above a speed of 180 revolutions per minute of worm-shaft which corresponded to a surface velocity of about sixty feet per minute at brake surface, these vibrations seldom occurred; but as the wheel slowed down they would almost invariably appear before coming to a full stop. Once they occurred at a speed of 348 revolutions after the wheel had become hot enough to melt the tallow, and their recurrence at this speed was prevented by keeping the wheel cool with water.

At another time, when running at a speed of seventy revolutions with the same load, the vibrations continued for several hours, and were finally checked by allowing the wheel to warm up. The speed was then reduced to three revolutions, at which it ran without shake or jar, but five minutes later, upon starting up, the vibrations were as severe as ever and could not be made to disappear, even at a speed of seventy revolutions.

The wooden blocks were then scraped and washed with benzine to remove all grease, with, it was thought, some improvememt, although at slow speeds, the vibrations still continued.

In general, it was found that slow speeds and heavy weights had the greatest tendency to produce vibrations, and that the heavier the load the greater the speed necessary to check them. At slow speeds their amplitude was greatest and their number the least, while as the speed increased, they became shorter and more rapid, producing a higher and higher tone until they finally ceased.

The phenomena presented were so strangely contradictory that for a time they seemed to defy explanation, and seriously to threaten the success of the undertaking.

The only plausible theory upon which they could be accounted for appeared to be, that the frictional surfaces were in an unstable condition, caused by the difference between friction of rest and friction of motion.

With most substances the friction of motion is less than the friction of rest, and it was argued, that by reason of the friction of rest, the brake strap would be carried beyond the point where it belonged, to be in equilibrium with the friction of motion, and when sliding occurred, it would fall back and grip the wheel with such force as to stop all sliding and again produce friction of rest.

Having failed to secure a satisfactory brake surface of wood it was decided to try another material, and from the view of the case just presented, leather suggested itself as the most suitable substance, on account of its peculiar frictional properties, which make it an exception to the general rule, that friction of motion is less than friction of rest.

Accordingly, each block of wood was covered with a leather face, and all further difficulties of this kind were effectually checked.

During the long series of experiments which were afterward made, a slight tremor was sometimes noticed when the wheel became hot enough to dry up the leather, but at such times the original condition was easily restored by the application of a moderate amount of belt grease.

Too much of the lubricant was found to be injurious, and no more was needed than could be absorbed by the leather. It was also noticed, that on account of the increase in friction of the leather over the wooden surface, the brake was more easily adjusted, and that the weight suspended from it hung, in consequence, farther from the centre of the drum shaft.

It could hardly be expected that a weight suspended in this manner from a rotating wheel would remain motionless, and very naturally it was found that oscillations of gradually increasing amplitude were to be constantly contended with. The disturbing cause, however, was so slight that the friction produced on the guide *O*, by a weight of one pound resting upon the graduated bar *M N*, was found to be sufficient to reduce the variation in its readings to one-sixteenth inch. The apparatus was now thought to be entirely satisfactory and the experiments were begun anew.

The readings of the graduated bar were taken to the nearest one-sixteenth of an inch, and they were considered accurate within that limit, which in a radius of fourteen inches, gives a probable error of less than ·005. During the progress of an experiment this radius was subject to gradual changes, which were generally accompanied by corresponding changes in the reading of the dynamometer.

The readings of the graduated bar and dynamometer were, however, always taken at the same instant, and as a matter of possible importance the temperature of the oil in the worm box was also recorded by a thermometer constantly immersed.

The dynamometer was sensitive to a variation of half a pound, and the readings for average cases were probably taken within ·01 of their true amount, the error in observation being greater for light loads and less for heavy ones.

It was not suspected until the experiments were about to be concluded, that the dynamometer itself was liable to any error at all, and it was then discovered, when too late, that it had a constant error of about two per cent., which was either positive or negative according to the alignment of the universal joints.

It will be explained further on how this error was detected and the causes which produced it, but in view of the great variations found in experiments made under similar conditions, it was not thought necessary to attempt to make any correction. Indeed, this could not have been done without a repetition of the whole work, and the experiments are accordingly presented as subject to an error of two per cent.

The probable errors in observation are not included, because it is not to be supposed that the average of so large a number of experiments could be much affected by errors in observation, unless

these errors were necessarily in one direction, as in the case just cited.

The merits and defects of the apparatus have been thus reviewed at length, partly for their own intrinsic worth, but chiefly as matters of the first importance to a clear appreciation of the real value of the experiments.

It is intended to give not merely the bare results, but also the facts upon which a judgment or criticism of them can be formed, and this is our apology for what may seem like unnecessary detail.

Having overcome the difficulties thus far enumerated the experiments themselves were continued.

Weights were suspended from the point *M*, ranging from 256 to 4,000 pounds, and the worm shaft was run under these loads at speeds ranging from three to 880 revolutions per minute. A great variety of conditions were thus obtained, and covered by about 800 experiments.

These conditions involved four variables, namely, speed, pressure, temperature, and state or nature of the rubbing surfaces.

The speed and pressure were the primary conditions adjusted by the operator, and the temperature and state of surfaces were secondary, and dependent upon the duration of the experiment as well as upon their primaries. The importance of the latter variable was not discovered until about half of the experiments were completed, and then it could be judged of only by its effects

The experiments have been divided into series, each representing a special set of gearing tested.

In every case the material used was cast iron, and to facilitate comparisons the wheels and pinions were all made as near as possible to the same dimensions.

The first series was made upon a double-thread worm, four-inch diameter, gearing with a worm wheel of thirty-nine teeth, one and one-half inch pitch, the thrust of the worm being taken on its annular surface instead of upon the step bearing used in other series. The remaining series upon this form of gearing were made upon worms having cast and cut teeth of single and double threads, all of the same pitch and diameter, gearing with the same diameter of wheel. Four series were made with spiral pinions four-inch diameter, one and one-half inch pitch, having respectively one, two, four and six teeth, gearing with a spur wheel of thirty teeth,

and one series with a spur pinion of twelve teeth, one and one-half inch pitch, gearing with the same wheel.

From the great mass of data obtained in this way, it became necessary to deduce some general conclusions by which the efficiency of any system of gearing could be determined.

The foundation for this work was taken to be the efficiency of the apparatus used. This was computed for every experiment by two persons independently, and their results were compared and corrected by a third, so that the possibility of errors in calculation should be reduced to a minimum.

The method of computation was very simple, but the labor involved by so many experiments was very great. The efficiency of the apparatus was in each case determined by dividing the moment of the dynamometer into the moment of resistance.

The moment of resistance was measured by the product of the weight suspended at M into the distance R, plus an additional amount for the moment of the brake itself, and the moment of the dynamometer was measured by the product of its record into the distance $A\ B$ times the ratio of the gearing used.

The efficiency was at once seen to depend principally upon the speed, and within limits, the higher the speed the greater the efficiency. But there were other disturbing elements, including temperature, pressure and state of surfaces, the combined effects of which produced many exceptions to this general rule.

At very high speeds, the rubbing surfaces were more liable to cut, and beyond certain points which could not be definitely determined, the efficiency appeared to diminish.

In different cases, variations of temperature and pressure were accompanied by such contradictory results that no generalizations could be made concerning them.

As already stated, the condition of the rubbing surfaces could be judged of only by its effect. At certain speeds and pressures the efficiency would slowly increase to its maximum, while at others it would suddenly diminish and indicate the destructive action known as cutting.

Upon examination this destructive condition did not always become apparent to the eye, and in some cases the apparatus was taken apart and cleaned, without making any decided improvement in efficiency, but it was finally discovered that in such cases

it was necessary, to restore the surfaces to their best condition by running for some time at a moderate speed and pressure. From this it appeared that the order in which the experiments were made, had an important bearing upon the results obtained, and it also furnished a satisfactory explanation for the contradictory appearance of many experiments which were otherwise made under apparently the same conditions.

Throughout one-half of these series no attention was paid to the order of the experiments, their sequence being guided entirely by convenience. The main object at first was to discover the efficiencies corresponding to variations in speed, pressure and temperature, and to determine a definite limit beyond which the speed of worm gearing could not be carried to advantage.

The injurious effects of high speeds upon succeeding experiments were not at first noticed, and a number of series are thus intermixed with exceptional cases arising from this cause, while in others the destructive conditions appear to form a separate group.

Having found the limiting speeds and pressures for worm gearing, care was taken in the remaining series to avoid them as far as possible, and by this means to secure some very good and harmonious results.

In order to present the experiments in a convenient shape for generalization and practical use, the attempt was made to tabulate them with reference to the speed as the most important variable. But the tabular method proved to be inadequate, unwieldy, and difficult of application to other cases, and in its place was substituted a graphical method, by which the efficiency and attendant conditions in each series could be seen in a much more comprehensive and instructive manner.

The results of each series herewith presented have been carefully plotted to scale, making abscissas proportional to the logarithms of the revolutions per minute and ordinates proportional to the efficiency of the apparatus.

This peculiar method of constructing the abscissas was adopted because it was seen that increments in efficiency were more nearly proportional to the ratio of the speeds than to their actual differences.

Different symbols are used to denote different pressures on teeth, and each symbol is numbered to show the order in which

the experiments were made and the temperature of the oil in the worm box.

These symbols are connected so as to form lines of efficiency corresponding to the various pressures used, and the average of all these lines is taken as the average for the whole series.

The first series of experiments graphically represented in *Fig. 90* was made upon a double-thread worm four inches diameter, gearing into a worm wheel of thirty-nine teeth one and one-half inch pitch.

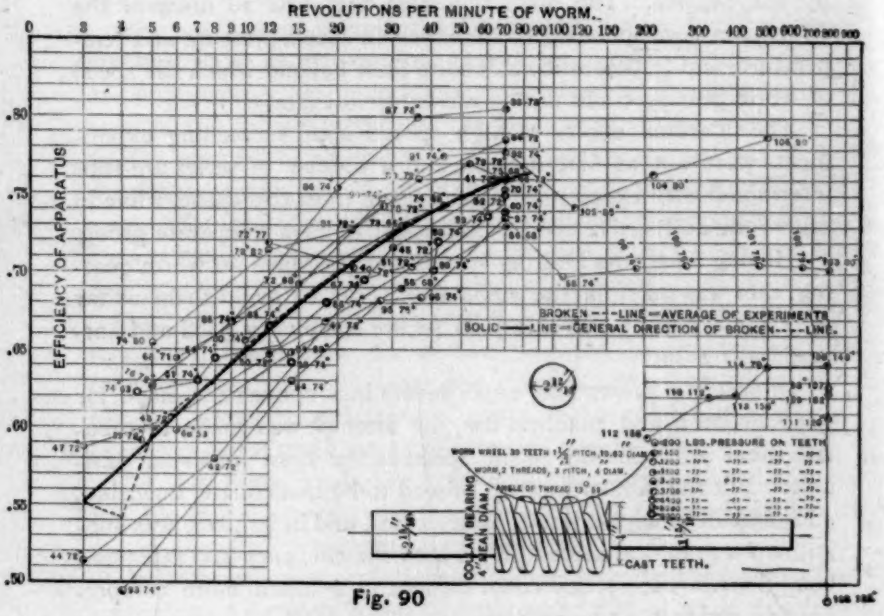

REVOLUTIONS PER MINUTE OF WORM.
EFFICIENCY OF APPARATUS
BROKEN - - -LINE—AVERAGE OF EXPERIMENTS
SOLID ——LINE—GENERAL DIRECTION OF BROKEN- - -LINE.
CAST TEETH.

Fig. 90

There are 114 experiments which cover a range of speed from three to 790 revolutions per minute, and loads up to 4,000 pounds upon brake wheel.

The worm and wheel had cast teeth, and the thrust of the worm was taken upon the worm itself instead of upon the shaft step, as in the subsequent series.

Of these 114 experiments the first thirty-eight have been omitted as preparatory on account of difficulties already stated, and the diagram begins with experiment thirty-nine, when the leather lagging on brake strap was first used.

The first sixty-two experiments were made with the disc dynamometer, and from there on, gear wheels were used to diminish journal friction.

The diagrams herewith presented give the essential results obtained, and it will only be necessary in connection with them to notice some peculiarities.

Considerable variation in the efficiency of the apparatus at any given speed will be observed on all diagrams, but more especially upon those for worm gearing, where, as already stated, no attention was paid to the progressive order of the speeds and loads.

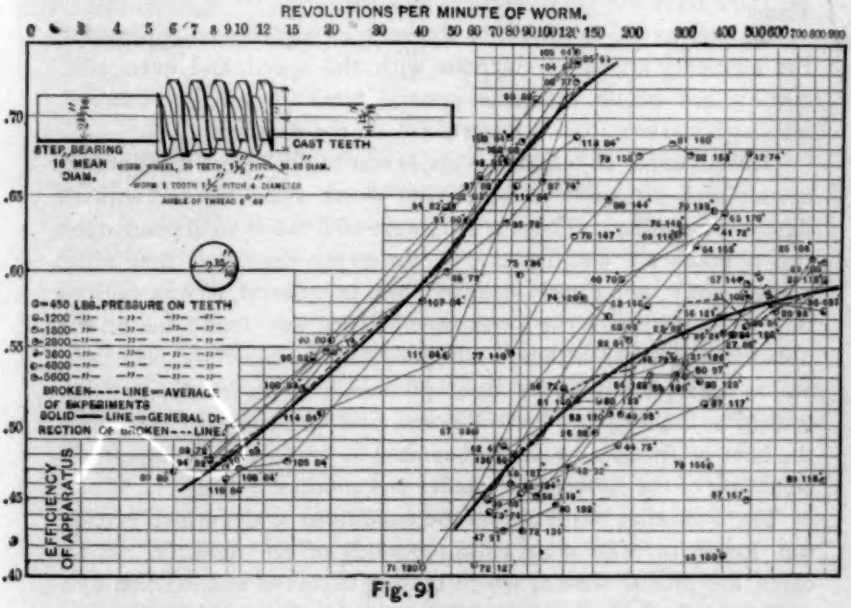

REVOLUTIONS PER MINUTE OF WORM.
0 2 3 4 5 6 7 8 9 10 12 15 20 30 40 50 60 70 80 90 100 120 150 200 300 400 500 600 700 800 900
.70
.65
.60
.55
.50
.45
.40
EFFICIENCY OF APPARATUS
STEP BEARING 16 MEAN DIAM.
CAST TEETH
BROKEN---LINE=AVERAGE OF EXPERIMENTS
SOLID——LINE=GENERAL DIRECTION OF BROKEN---LINE

Fig. 91

In *Fig. 90*, the bulk of the experiments were made between the limits of three and seventy revolutions per minute, for, in order to obtain the higher speeds, it was necessary to make a change of pulleys, and, fortunately, this was not done until the experiments on the lower speeds were completed.

Between the limits just mentioned, the general direction of the lines of efficiency is strongly marked, and the group forms a band of nearly uniform width, extending over about ten per cent. of variation. At the higher speeds the results are very much scattered and give unmistakable evidence of cutting.

This series was unexpectedly brought to a close by the breaking of the worm teeth, which were found to have been entirely ground away. The wheel, however, was in good condition, the wear having been just sufficient to polish the surface of the teeth.

The second series of experiments was made upon a single thread cast worm and wheel, the thrust being taken on the step at end of worm shaft. In this series, also, the experiments appear to form themselves into separate groups, and the same phenomenon will be noticed on *Figs. 92*, *93* and *98*, and in all cases where the destructive conditions depending upon speed, pressure and temperature have not been carefully avoided.

So long as the condition of the rubbing surfaces is unimpaired, the efficiency appears to increase with the speed, and even after cutting has begun, the same general tendency will be observed, although starting from a lower point on the diagram.

The principal object in this series was to determine the limiting speeds and pressures at which the worm could be run without danger of cutting. These speeds were also found to depend upon the temperature and duration of the experiment, and they were, consequently, not clearly defined; but, in general, it was noticed that at slow speeds the greatest efficiency was found under the heaviest pressures, at moderate speeds under moderate pressures, and at high speeds under light pressures. This seemed to point toward a limiting product of speed and pressure, beyond which the heat of friction became generated so rapidly as to impair the condition of the surfaces in contact and produce cutting.

The following table shows the conditions under which cutting was found to take place within periods of ten minutes. In all cases, the precise time at which cutting occurred was marked by a sudden rise in the reading of the dynamometer, and from the reading at that time the final efficiencies were computed. In the last experiment no cutting took place, as shown by the constant efficiency of ·677.

SPEEDS AND PRESSURES LIABLE TO PRODUCE CUTTING.

Velocity of Sliding in Feet per Minute.	Pressure on Teeth.	Temperature.		Efficiency.		Duration of Run in Minutes.	Foot-Pounds per Minute consumed in Friction before Cutting began.
		Initial.	Final.	Initial.	Final.		
800	1785	106°	140°	·609	·387	6	117,600
880	1780	118°	132°	·607	·462	3	129,300
880	1205	137°	150°	·575	·360	3	97,000
800	448	118°	133°	·594	·445	10	29,400
480	2822	144°	167°	·591	·450	7	117,800
400	3481	170°	180°	·639	·415	3	98,300
360	4837	138°	166°	·641	·473	6	122,400
306	5558	163°	186°	·677	·677	10	102,000

It appears, from this table, that the danger of cutting does not depend entirely upon the amount of frictional work, and it is not easy to understand why this should be the case, but the fact remains, and we are forced to conclude that very high speeds should be avoided, even under light pressures, and that the best working conditions are to be looked for at or below 300 revolutions per minute, which corresponds to a surface velocity of sliding of about 300 feet per minute. A great deal, of course, depends upon the rapidity with which the heat of friction can be conducted away.

For the apparatus used, it was determined that when one horse-power was consumed in friction, the worm-box remained at a uniform temperature of about 50° above the surrounding air, and, assuming the rate of cooling to be proportional to the difference in temperature, it seems improbable that more than two horse-power could be continuously consumed in friction without overheating the lubricant.

In plotting the diagrams care has been taken to exclude from the general result the worst and most evident cases of cutting, such as those just given in the table.

The third series, represented in *Fig. 92*, was made on gearing similar to that used in the first series, with the exception that the thrust of the worm was taken upon the step at end of worm shaft instead of upon the annular end of the worm itself. At the conclusion of this series, the step bearing was found to be in good

condition, the rubbing surfaces having worn to an annular bearing of, say from one to three inches diameter.

The worm shaft and its bearing at large end, however, were badly cut, and, judging from the record of the experiments, the cutting must have occurred near the middle of the series, thereby reducing the efficiency of the latter half of the experiments, and furnishing an explanation for the discrepancies between this and the first series. In this series the limiting pressure for a speed

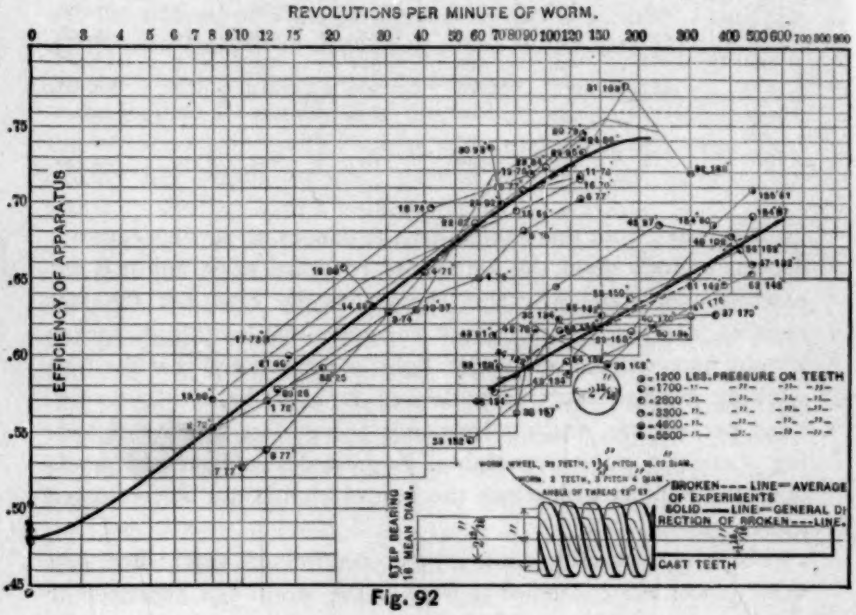

REVOLUTIONS PER MINUTE OF WORM.
EFFICIENCY OF APPARATUS
1200 LBS. PRESSURE ON TEETH
BROKEN --- LINE = AVERAGE OF EXPERIMENTS
SOLID —— LINE = GENERAL DIRECTION OF BROKEN --- LINE.
STEP BEARING 1⅝ MEAN DIAM.
CAST TEETH

Fig. 92

of 300 revolutions appeared to be about 4,500 pounds for a run of five minutes.

A pressure of 5,600 pounds at a speed of 280 revolutions per minute and temperature of 190° produced cutting in three minutes, but whether on the teeth or in the journals it was never possible to determine without taking the apparatus apart at the sacrifice of considerable time. But this cutting would naturally be supposed to take place sooner on the teeth where the intensity of pressure was greater.

One more series, the ninth in order, represented in *Fig. 98*, completes the experiments upon worm gearing. This was made

upon a worm and wheel similar in every respect to those used in the second series, except that the teeth were cut instead of being cast. The speeds, however, were kept below the limits found by previous experiments to produce cutting, and the temperatures were consequently moderate, ranging from 48° to 116° in extreme cases. In this way the rubbing surfaces were kept in good condition throughout, and the results are altogether more satisfactory than any yet recorded.

The break in the line of efficiency at the higher speeds is not due in this series to their injurious effect, but to the fact that the precaution of running the apparatus for some time before commencing to note experiments was accidentally neglected.

It had been previously observed that under moderate speeds and loads, the efficiency of the apparatus would continue to improve until a maximum limit appeared to be reached, and, in order to obtain observations under the best working conditions, a rule had become established not to begin noting down experiments until after the apparatus had been warmed up once or twice in running.

For all speeds below 100 revolutions, however, the curve shown on *Fig. 98* may be accepted as very near the truth, and, by comparing this with *Fig. 91*, it seems probable that with both sets of gearing in their best working condition, these lines of efficiency would practically coincide.

From the experiments made to determine the limiting speed and pressure at which worm gearing can be run successfully, we conclude that when the gearing is loaded to its working strength, it is not safe to exceed a velocity of sliding of 300 feet per minute, and that, in general, the best working conditions are obtained at or about a velocity of 200 feet per minute. The experiments upon spiral gearing begin with *Fig. 93*, which shows the efficiency of a one-toothed pinion gearing with a spur wheel of thirty-nine teeth one and one-half inch pitch.

In these experiments, the shaft *G H* was set around at an angle of 6° 51′, so that the teeth of the spur wheel should gear properly with the pinion. The results obtained show a constant improvement in the condition of the rubbing surfaces throughout the series, and clearly indicate the necessity just mentioned, of running the apparatus for some time before taking observations. Here the

slow speeds were taken first before the teeth were worn to a good bearing, while in the ninth series the circumstances were reversed. In both series the condition of the surfaces underwent considerable change, and the results are to that extent not as satisfactory as could be desired. Upon repeating some experiments at the conclusion of this series, it was noticed that the efficiency had improved nearly ten per cent., and the broken line on the diagram

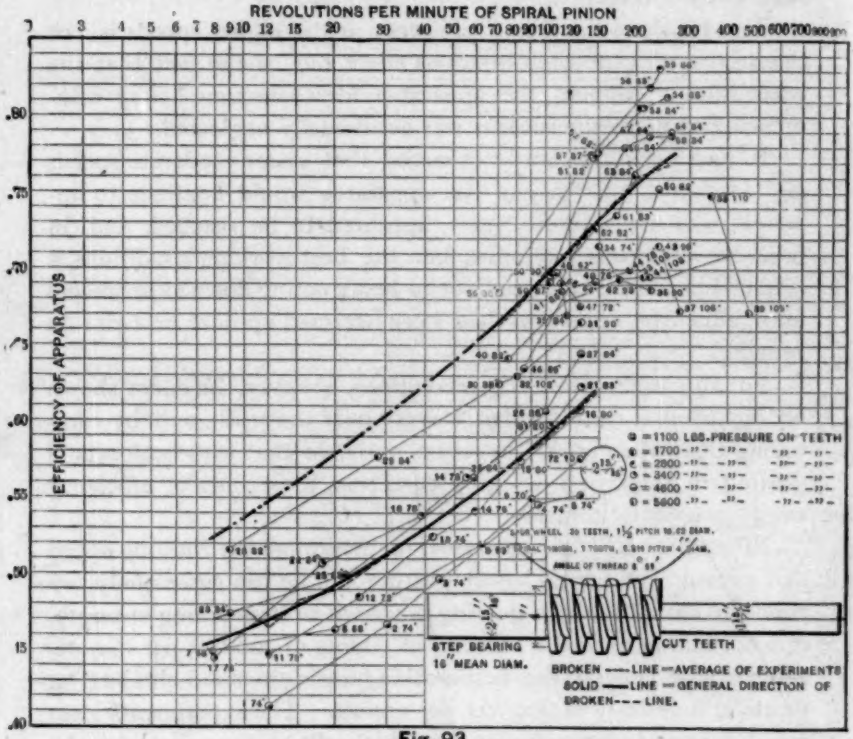


Fig. 93

was drawn to show the probable position of the curve, had the precautions alluded to been observed. Theoretically, the efficiency of the spiral gear should have been a trifle better than that of worm, on account of a slight diminution in the velocity of sliding, and had this series been repeated, it would undoubtedly have so appeared. The teeth used in this gearing were accurately cut to an epicycloidal shape, and in all cases their action in rolling contact was practically perfect, while with the worm gearing the shape

was rather indefinite, presumably involute, but the patterns for the wheels were made by a cutter, like the worm itself, and the two were necessarily obliged to fit.

The fifth series was made upon a spiral pinion of four teeth, with the shaft *G H* set around at an angle of 28° 31′. In this, as well as the following series of experiments upon spiral pinions, the apparatus was run at a moderate speed and load, until the rubbing surfaces were thought to be in their best working condition, before any records were taken, and as a result, the diagrams will be found

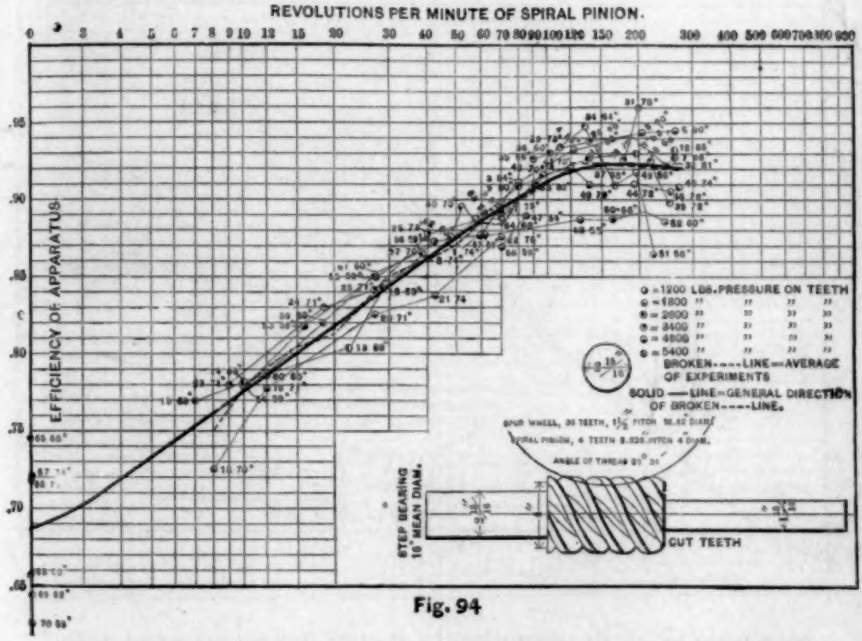

REVOLUTIONS PER MINUTE OF SPIRAL PINION.
EFFICIENCY OF APPARATUS.
Q = 1200 LBS. PRESSURE ON TEETH
BROKEN ---- LINE = AVERAGE OF EXPERIMENTS
SOLID — LINE = GENERAL DIRECTION OF BROKEN ---- LINE.
CUT TEETH

Fig. 94

to be more definite and conclusive than anything heretofore shown. The speeds ran just high enough to show that the limit had been nearly reached, but no well-defined case of cutting occurred, and the experiments may be regarded as having been taken under the best working conditions.

The sixth series was made upon a spiral pinion of two teeth, with the shaft *G H* set around at an angle of 13° 49′. Here also the results obtained are remarkably progressive, and free from all evidence of cutting. A comparison with similar experiments upon

the double-thread worm shows a great gain in efficiency—which cannot be credited entirely to the improved action of the gear teeth, but which must also be accounted for by the fact that destructive speeds were avoided, and that the apparatus was maintained throughout in good working condition.

The seventh series, upon a spiral pinion of six teeth with the shaft *G H* set at an angle of 45° 44′, also gives very good results, and these three series, *Figs. 94, 95* and *96*, might be taken as a

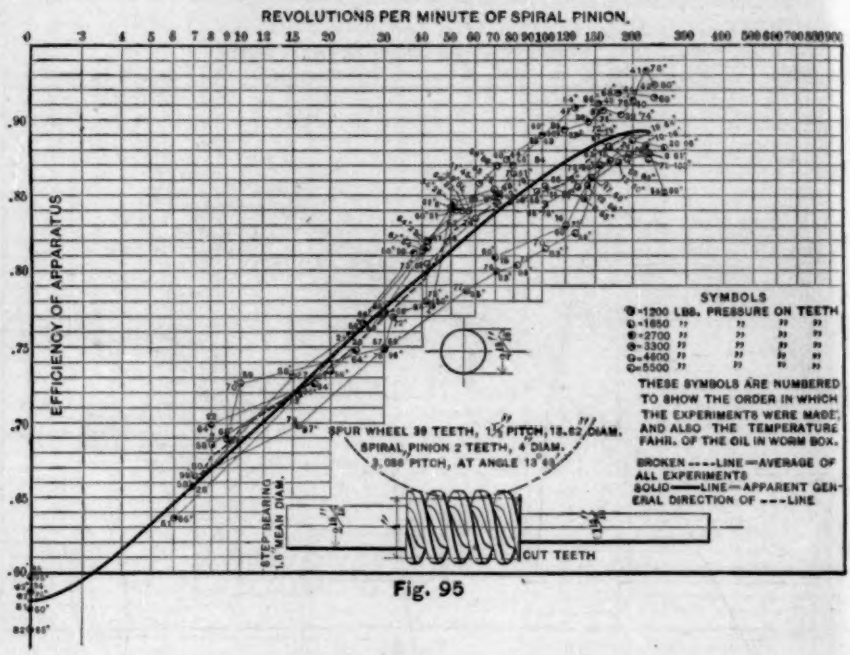

REVOLUTIONS PER MINUTE OF SPIRAL PINION.
EFFICIENCY OF APPARATUS
SYMBOLS
THESE SYMBOLS ARE NUMBERED TO SHOW THE ORDER IN WHICH THE EXPERIMENTS WERE MADE, AND ALSO THE TEMPERATURE FAHR. OF THE OIL IN WORM BOX.
BROKEN ----LINE—AVERAGE OF ALL EXPERIMENTS
SOLID——LINE—APPARENT GENERAL DIRECTION OF ---LINE
SPUR WHEEL 30 TEETH, 1½ PITCH, 18.62 DIAM.
SPIRAL PINION 2 TEETH, 4 DIAM.
3.088 PITCH, AT ANGLE 13° 03′
STEP BEARING
CUT TEETH

Fig. 95

basis upon which to determine the laws of friction for spiral gearing in general.

To make a complete analysis of these results, it would be necessary to eliminate the friction of worm shaft and drum shaft from that of the teeth themselves, and determine the coefficient of friction for a number of different velocities of sliding.

An effort was made to do this by running these shafts under known journal pressures, but the results obtained were of such an indefinite and contradictory character that the task seemed hopeless, and it was not until after the completion of the eighth series

that a method was adopted by which the efficiency of any set of gearing could be approximately determined. For practical convenience the efficiency of the whole apparatus may be considered as equal to that of any other similar piece of gearing, and, as a matter of fact, the relative dimensions of the various parts do not differ a great deal from those used in these experiments.

The wheels might be larger or smaller, but the diameters of spiral pinions and worms generally bear about the same relation to their shafts, and it is in these parts that the principal work of friction is consumed. When the angle of a worm or pinion and

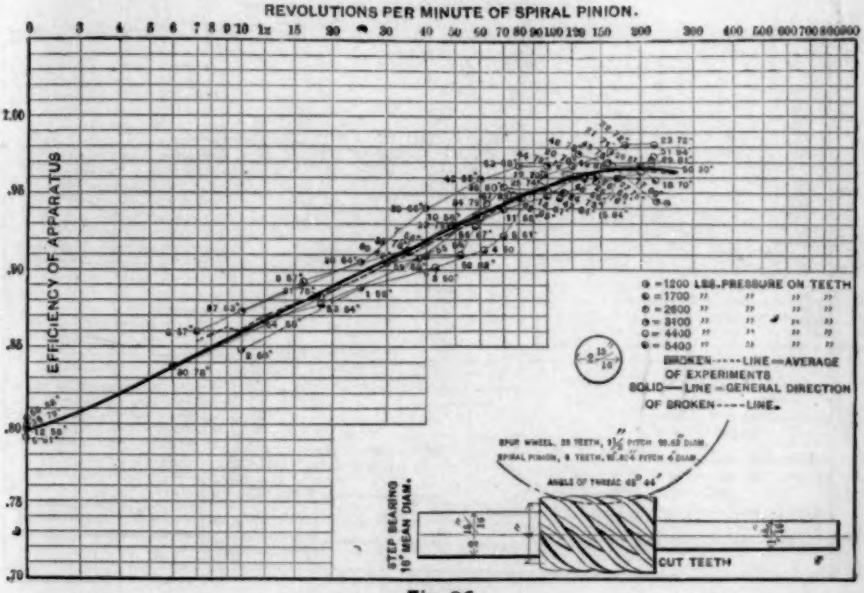

REVOLUTIONS PER MINUTE OF SPIRAL PINION.
EFFICIENCY OF APPARATUS
1.00
.95
.90
.85
.80
.75
.70
W = 1200 LBS. PRESSURE ON TEETH
W = 1700 " " " "
W = 2600 " " " "
W = 3800 " " " "
W = 4400 " " " "
W = 5400 " " " "
BROKEN-----LINE = AVERAGE OF EXPERIMENTS
SOLID—— LINE = GENERAL DIRECTION OF BROKEN----LINE.
STEP BEARING 10" MEAN DIAM.
CUT TEETH

Fig. 96

its speed are given, it should be possible, from these experiments, to fix at once with tolerable accuracy the efficiency of the gearing to which it belongs.

This can now be done for the angles experimented upon, but we shall presently show how, by the method alluded to, the efficiency may be determined for any angle whatever.

The eighth series (*Fig. 97*) was made upon a cut spur pinion of twelve teeth one and one-half inch pitch, with the shaft *G H* set parallel to the pinion shaft. In this series an error, already mentioned, was discovered in the readings of the dynamometer. The

efficiencies, as calculated from the record of experiments, were in some cases found to exceed 100 per cent., and these results were repeatedly confirmed by experiments.

Numerous efforts were made to locate the error without success, until finally it was remembered that the accuracy of the instrument was predicated upon the assumed flexibility of the universal joints. It was also thought probable that however carefully adjusted the shafts *F E* and *C D* might be, they would no doubt be somewhat out of line, and that consequently the universal joints would be forced to swivel slightly under pressure. Should the

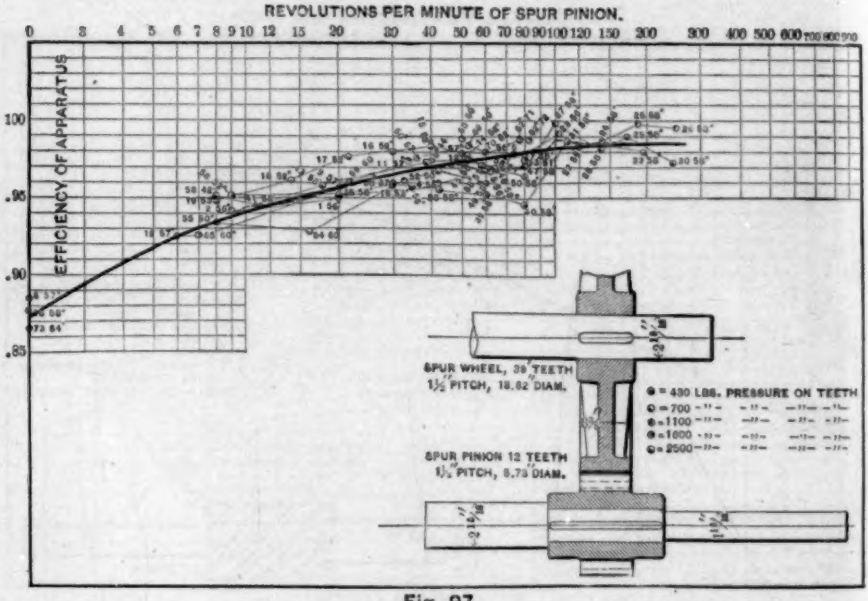

REVOLUTIONS PER MINUTE OF SPUR PINION.
0 3 4 5 6 7 8 9 10 12 15 20 30 40 50 60 70 80 90 100 120 150 200 300 400 500 600
EFFICIENCY OF APPARATUS
100
.95
.90
.85
SPUR WHEEL, 36 TEETH
1½ PITCH, 18.62 DIAM.
SPUR PINION 12 TEETH
1½ PITCH, 6.75 DIAM.
● = 430 LBS. PRESSURE ON TEETH

Fig. 97

error be due to this cause it was expected that by moving the shaft *F E* parallel to itself for a short distance to either side of the centre line of *C D*, the efficiency would appear to be alternately above and below its true value by an amount due to the stiffness of these joints.

The shaft *E F* was accordingly moved about one inch from the centre line in either direction and a number of experiments were made in each position, the result of which showed a total variation of about ·04 in the average efficiencies as determined for each position and speed.

For a speed of twenty revolutions the efficiency varied from ·944 to ·993 according to the position of *E F*, and the true value was accordingly estimated at ·968, the arithmetical mean. Similarly the efficiencies at a speed of eighty revolutions, varied from ·966 to 1·016, giving ·991 for the true value at this speed.

The eighth series, as plotted upon *Fig. 97*, represents the mean of two sets of experiments conducted in this way, and the errors due to the stiffness of the universal joints are consequently eliminated. It should be observed in regard to this error, that its

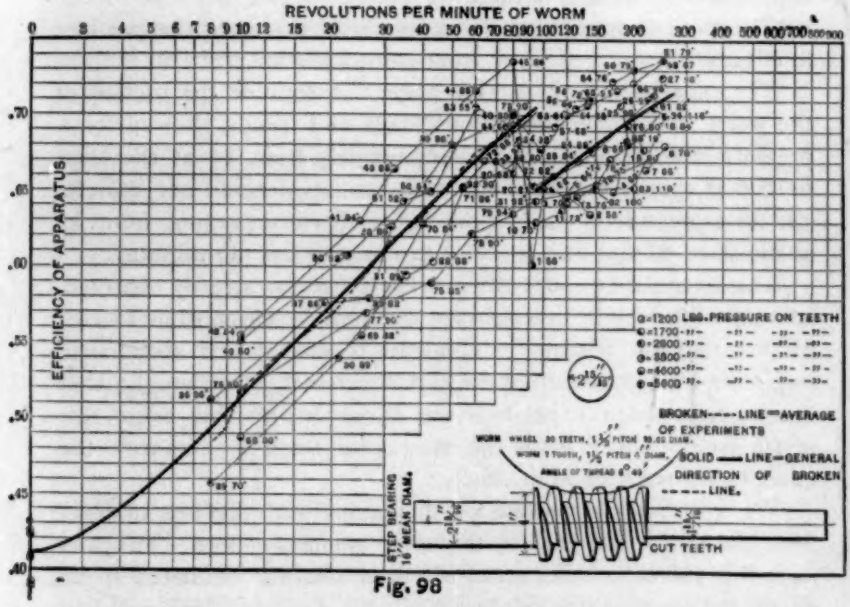

REVOLUTIONS PER MINUTE OF WORM
EFFICIENCY OF APPARATUS
BROKEN---- LINE = AVERAGE OF EXPERIMENTS
SOLID —— LINE = GENERAL DIRECTION OF BROKEN ---- LINE.
CUT TEETH

Fig. 98

actual amount is proportional to the efficiency itself, or, in other words, that the error is two per cent. of the efficiency and not an actual difference of two per cent. more or less, and for this reason it may be said there was less necessity for its correction in any other series than there was in this. An approximate analysis of the loss in effect presents fewer difficulties in this series than in any other, but still it is necessary to make certain hypothetical assumptions which cannot as yet be substantiated. This loss in effect may be considered as composed of three parts, the friction of two journals and the intervening gear teeth.

If we assume the loss in each case to be proportional to the product of the pressure and amount of sliding, we have the pressure on pinion shaft equal to the pressure on teeth, and the pressure on drum shaft about fifty per cent. greater, owing to the resultant from load suspended.

The amount of sliding per revolution of pinion shaft is about 7·5 on pinion shaft, 3 inches on teeth, and 3 inches on drum shaft, and the product of these numbers times the pressure in each case gives the relation of 7·5, 3, and 4·5, the sum of which is 15. Of this total effect the friction of the gear teeth consumes ·2.

Therefore, it appears on this assumption, that the principal loss in effect with spur gearing is from journal friction.

The experiments which were made to determine the friction of the worm shaft under the various loads and speeds, showed for a pressure of 800 pounds, coefficients ranging from ·15 at the start to ·006 at speeds between 110 and 240 revolutions per minute, and for a pressure of 3,440 pounds, coefficients ranging from ·12 at the start, to ·033 at a speed of 240 revolutions per minute.

This irregularity under different loads made, as already observed, so much difficulty in determining the loss in effect from journal friction, that the attempt to eliminate it was at first abandoned, but, after the completion of the eighth series it was thought that inasmuch as the principal loss was there due to this cause, this series itself might be made the basis upon which to eliminate the effect of journal friction in others.

We will therefore assume for this series that the loss in effect from journal friction is ·8 of the total amount of loss. The loss from this source in other series has been roughly estimated to be nearly the same, and proceeding from this stand-point we can find the probable coefficients of friction for the pinions used in the fifth, sixth, and seventh series.

The efficiency of the apparatus, E, plus L, the loss in effect from journal friction gives the efficiency of the spiral pinion and its step. From this we can find the coefficient of friction by assuming the coefficient for step friction to be the same as for the teeth themselves.

Let α = angular pitch of spiral pinion.

φ = coefficient of friction.

n = ratio of mean diameter of step to pitch diameter of pinion = ·4.

Then it can readily be shown that

$$E + L = \frac{1}{1 + (1 + n)\,\varphi \cot a} \tag{1}$$

whence

$$\varphi = \frac{1 - (E + L)}{(E + L)(1 + n) \cot a} \tag{2}$$

From these formulæ a number of coefficients have been calculated, and their values given in the following table, from which it will be seen that the coefficients agree as closely as could be expected, and thereby indicate to a certain extent an harmonious relation between the curves employed in their determination.

Revs. of Pinion.	E	L	E			ϕ			Average Value of ϕ.
	8th Series.		5th Series 4 Teeth.	6th Series 2 Teeth.	7th Series 6 Teeth.	5th Series 4 Teeth.	6th Series 2 Teeth.	7th Series 6 Teeth.	
3	·90	·08	·70	·59	·81	·105	·086	·094	·095
5	·92	·064	·73	·63	·83	·097	·078	·089	·088
10	·94	·048	·774	·685	·86	·081	·064	·076	·074
20	956	·035	·817	·741	·89	·065	·050	·061	·059
50	·973	·022	·876	·813	·93	·042	·035	·038	·038
100	·982	·014	·912	·862	·955	·030	·025	·024	·026
200	·984	·013	·923	·892	·967	·026	·018	·015	·020

The discrepancies are really not so great as they appear when we consider the variations in the experiments themselves, and the practical impossibility of obtaining very accurate results.

The average of these coefficients, it was thought, would represent, as nearly as possible, the best general results of the whole course of experiments; but upon still further comparison with the best parts of the other series, it was concluded that the sixth series itself formed a better standard from which to deduce hypothetical curves of efficiency for other pinions not included in the experiments. In this series, the probable error in these coefficients was less than in either of the others, because the pinion and step consumed a larger proportion of the total loss in effect, and had the series upon the single tooth pinion been made with equal care, the error would have become still less.

We are well aware that the method just employed for the determination of these coefficients is open to criticism, but when judged by the character of the data with which we had to deal, we believe it to be as refined as the conditions would permit, and however faulty it may be in theory, it is evident that by reversing the process we can reproduce from these coefficients the original curves from which they were deduced.

The effort has been to find some law, by analysis if possible, or by trial, if necessary, which would cover the best portions of all the experiments. In this we have failed to obtain sufficiently accurate data for a complete analysis of the problem, and we have reluctantly been obliged to feel our way by the aid of assumptions whose merits were largely founded upon the results which they afterward brought about.

We have assumed, for instance, that the loss in effect from journal friction is the same for all series, that the coefficients for different portions of the apparatus are alike, and that the velocities of sliding for the different pinions were equal, to all of which exceptions might very properly be taken, but the results obtained upon these assumptions have agreed so well with each other that, for the present, we are obliged to accept them as correct, at the same time hoping that there may be enough interest awakened in the subject to lead to a better understanding of it in the future.

In order to put the results of these experiments in the best shape for practical use, a number of curves have been computed and plotted in *Fig. 99*, from which it is easy to find by interpolation the efficiency corresponding to any angle and speed of pinion.

To illustrate the practical use of this diagram, let us consider the following train of gearing and proceed to find its efficiency. Given a spur pinion driving a spur wheel, upon the shaft of which is a spiral pinion of 30° angle, driving another spur wheel upon whose shaft is a worm of 7° angle, driving a worm wheel.

When the speed of the first shaft is known, it is easy to find from the dimensions and ratio of the gearing, the velocity at pitch line of each pinion or worm. We will assume 200 feet per minute for the spur pinion, 50 feet per minute for the spiral pinion, and 10 feet per minute for the worm, and by reference to the diagram we find ·985 for the efficiency of the spur pinion ·902 for that of the spiral pinion and ·53 for that of the worm.

The efficiency of this train of gearing is therefore

$$\cdot 985 \times \cdot 902 \times \cdot 53 = \cdot 471.$$

For speeds above 200 feet per minute it is recommended to estimate upon the efficiency at that speed.

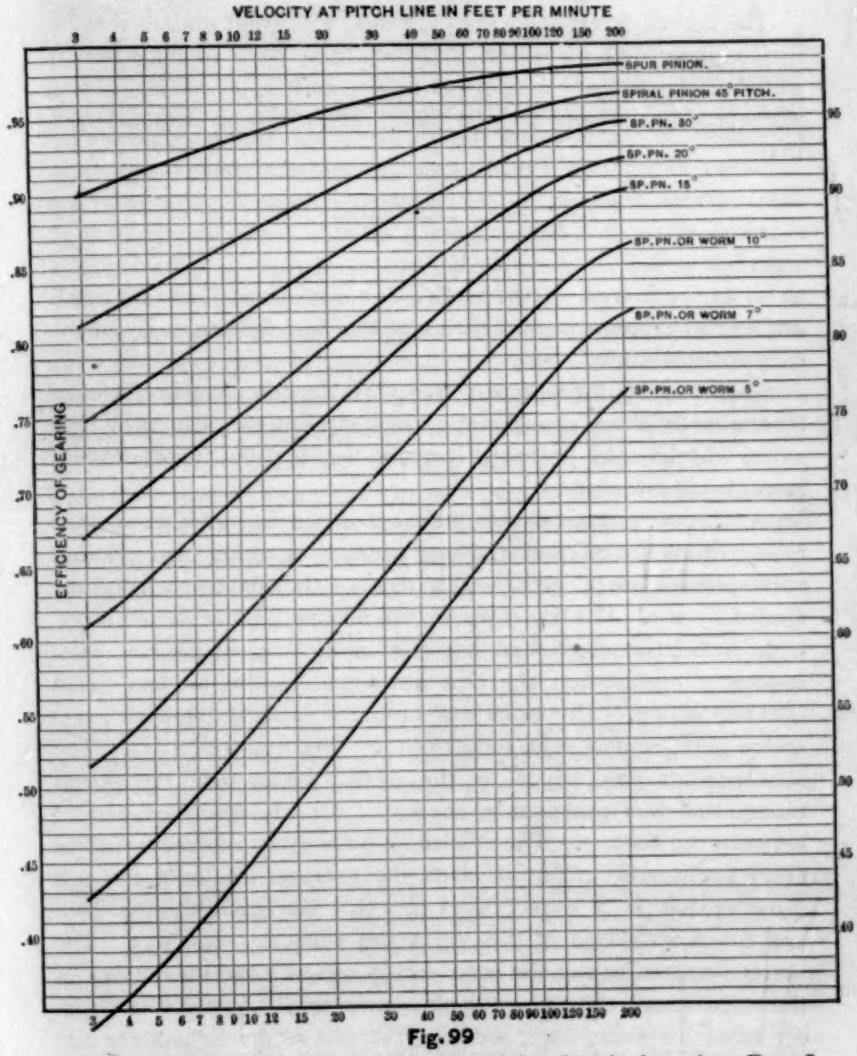


Fig. 99

From the coefficients as determined for the sixth series, $E + L$ can be found by substitution in the formula.

$$E + L = \frac{1}{1 + 1{\cdot}4\ \varphi \cot a}$$

and the value of E is then easily determined by subtracting the value of L, as just given in the table.

The curves determined for 5°, 7° and 10° spiral pinions may be used equally well for worms, and they have been so marked. If we let E = efficiency of a spiral pinion, and E_1 = efficiency of a worm having the same angular pitch, then it can be demonstrated that

$$\frac{E}{E_1} = \frac{\cot a}{\cot a - \varphi}$$

as far as the friction of the teeth alone is concerned, and for small angles this relation is so near unity that the difference is practically inappreciable.

Theoretically, for worms of 20° pitch, the difference in efficiency for the teeth alone should not exceed ·03, but the side thrust on worm wheel would probably increase the friction considerably if larger angles were used.

Additional experiments, not shown on the diagrams, were made to determine the efficiency of spur gearing, in which the reduction corresponded more nearly with that of the single and double-thread worms. For this purpose, the driving gear of an old forty-eight-inch lathe which had been for many years in use, and consequently represented the best attainable conditions, was called into requisition. This lathe was provided with double-back gears which gave, respectively, reductions of 38·4 and 10·2 to 1. The counter-shaft was driven by the dynamometer, and the power transmitted was measured by the brake, already described, carried between the centres. The efficiency of the lathe was measured for every speed under different loads, the average results of the test showing that for a reduction of 10·2 to 1 the efficiency was ·952, and for a reduction of 38·4 to 1 the efficiency was ·935. The power required to run the lathe empty was, of course, excluded in the calculations, otherwise the efficiencies would have had a different value for every load, becoming almost *nil* for light loads and approaching the figures given for heavy ones such as the lathe was expected to carry.

These experiments showed conclusively the advantage of spur gearing over all other kinds in point of efficiency and durability, for not only is there less friction, and consequently less wear, but the wear in spur gearing is distributed over a great deal more surface.

In conclusion it may be said, that the whole subject of friction is dependent upon such a variety of changeable conditions that it is almost impossible to separate and determine the effect of each independent variable apart from others which attend it. The four variables—speed, pressure, temperature and condition of surface—are susceptible of infinite combinations, and even if it were possible to measure the effect of each independently, it is quite probable that the effect of any given combination would not follow from that of each component factor.

To give some idea of the extent of these experiments and the time and care spent in arriving at the results herewith presented, it may be stated that the experiments began on the 8th of August, 1883, and were not completed until January 14, 1884, that fully six months were consumed in the design and construction of the apparatus, and as many more in working up the records obtained.

That they were not more completely successful is due to the fact that we attempted too much at first, and only learned by experience the influences which operated to impair the results.

AURORA SOUNDS.—In March, 1885, Sophus Tromholt despatched some thousand circulars to all parts of Norway containing different queries regarding the aurora, and amongst them also the following: "Have you or your acquaintances ever heard any sound during aurora, and, in this case, when and in what manner?" Up to September 16th, he had received answers to these queries from 144 persons. Of these, not less than ninety-two, or sixty-four per cent., believe in the existence of the aurora-sound, and fifty-three (thirty-six per cent.) state that they have heard it themselves, whilst the other thirty-nine cite testimonials from other people; only twenty-one (fifteen per cent.) declare that they have never heard the sound and know nothing about it, and the other thirty-one (twenty-two per cent.) have not noticed the query at all. There are thus ninety-two affirmations against twenty-one negations. The sound is variously described in these answers as sizzling, creaking, whizzing, rustling, crackling, hissing, whispering, rushing, buzzing, rippling, roaring, din, breezy, whipping, fanning, clashing, flapping, sweeping, etc.—*Nature, Sept. 24, 1885.* C.

THE DRAWING SCHOOL.

The usual closing exercises of the Drawing School were held in the Lecture room, on Thursday evening, May 13, 1886. The President, COL. CHARLES H. BANES, occupied the Chair. Addresses were made by the President, the Director and Prof. MacAllister, Superintendent of Public Schools. Numerous drawings of the pupils were on exhibition, and at the close of the addresses, diplomas were granted to students who had completed the full course of instruction. The following report of the Director, Mr. Wm. H. Thorne, was read, showing the operations of the School during the past year: (W.)

ANNUAL REPORT OF THE DIRECTOR OF THE DRAWING SCHOOL OF THE FRANKLIN INSTITUTE, FOR THE SESSIONS 1885–1886.

Notwithstanding the continued depression in manufacturing, this has been an eminently successful season of the Drawing School, in number of pupils, completeness of organization and efficiency of instruction. The number of pupils has been exceeded but once in the history of the school, and that was during 1882–3, a period of exceptional activity. The most encouraging feature of the season just passed has been the large attendance during the Spring Term, which was ninety-three per cent. of that during the Winter Term. The system of instruction in MECHANICAL DRAWING has been further improved, and, for correctness and thoroughness, is believed to be unexcelled. This department is under the immediate charge of the Director, MR. WILLIAM H. THORNE, and consists of JUNIOR CLASSES, under MR. WILLIS H. GROAT, for the study of Geometrical Problems and Elementary Projections; INTERMEDIATE CLASSES, under MR. CARL BARTH, for the study of Projections, Intersections and Developments; and a SENIOR CLASS, under the Director, for Technical Drawing. The ARCHITECTURAL and FREE-HAND CLASSES, under MR. CLEMENT REMINGTON, have made rapid progress and have done excellent work in making Plans, Elevations and Details of modern cottages in the same manner as in Architects' offices, and in copying from the flat in pencil and crayon, and in drawing in crayon and painting in oil from casts. These classes have been larger and more interested than ever before.

The students of the school are to be congratulated on the opportunity which has been given them, at a nominal cost, for obtaining a good foundation in the principles and practice of constructive and artistic drawing, and, as a rule, they have taken proper advantage of it.

Among them, the following deserve HONORABLE MENTION for their attention, industry and progress:

IN THE SENIOR MECHANICAL CLASS.

Thomas B. Main,
Joseph Hagman,
Ellwood M. Rowand,
Samuel Davis,
Barclay Thorn,
John Rowland.

IN THE INTERMEDIATE CLASSES.

Thomas Stephen,
Gilbert B. Downes,
T. Edward Schiedt,
William F. Stroud,
Sören Olsen,
Albert E. Atkinson,
Edward A. Muir.

IN THE JUNIOR CLASSES.

Frank R. Vernon,
W. G. Goodwin,
Henry Spiegel,
Theodore Fisher,
Leon Lake.

IN THE ARCHITECTURAL CLASS.

Horace T. Hatton,
Joseph Trottmann,
Arthur Cryer,
H. J. Parrott,
Charles M. Williams.

IN THE FREE-HAND CLASS.

Adolph Schmid,
William H. C. Swain,
John R. Griffiths,
Edward Winkens,
William Rothfuss,
Mary H. Goudkop,
Eugene Dillon,
Arthur Taylor,
Charles Scherer,
Edward Seatier.

The following students are awarded SCHOLARSHIPS from the B. H. BARTOL fund, entitling them to free attendance during the next Term, beginning SEPTEMBER 28, 1886, when they will present themselves to the Actuary and receive their tickets:

Thomas B. Main,
Thomas Stephen,
Frank R. Vernon,
Adolph Schmid,
William H. C. Swain,
Horace T. Hatton.

The following students, having attended the school during four terms, are awarded certificates to that effect:

Ellwood M. Rowand,	E. Morgan Denney,
John Rowland,	Franklin Farley,
William C. Bolgiano,	Thomas W. Draper,
William Hoeller,	Charles M. Williams,
Thomas Kirk,	William Rothfuss,
Edwin J. Rooksby,	Joseph Trottmann,
William T. McAllister,	Minnie M. Parker,

Guido Ferrari.

Continued efforts will be made to maintain and improve the high standard of the school, and to add to the usefulness and importance of this department of the FRANKLIN INSTITUTE.

WM. H. THORNE, *Director.*

Franklin Institute.

[*Proceedings of the Stated Meeting, held Wednesday, April 21, 1886.*]

HALL OF THE INSTITUTE, April 21, 1886.

COL. CHAS. H. BANES, President, in the Chair.

Present—111 members and thirteen visitors.

Eleven persons were reported to have been elected to membership since the last meeting.

The Chairman of the Committee on Science and the Arts reported that the Committee had recommended the award of the John Scott Legacy Premium and Medal to Dr. FRANCIS V. GREENE, of Philadelphia, for his process of Extracting Oil and Albuminoid Matter from Corn, etc.; and to LUCIUS J. PHELPS, of New York, for his System of Induction Telegraphy.

It was ordered that the Committee's recommendations in the cases above named be approved; and the Secretary was directed to communicate the action of the INSTITUTE to the Committee on Minor Trusts of the Board of City Trusts.

The Special Committee on Reorganization made a lengthy report which was accepted, and the Committee continued. It was further resolved that the Committee be directed to prepare, in accordance with the present By-Laws, the necessary changes in the same in order to carry into effect the recommendations of the report; and that the Committee be requested to prepare plans for a new building, and select a suitable site for the same. It was also ordered that the Committee's report, together with the text of the proposed amendments, be printed and a copy sent to each member of the INSTITUTE.

The Secretary presented his monthly report of progress in Science and the useful Arts.

Adjourned.

WM. H. WAHL, *Secretary.*

[*Proceedings of the Stated Meeting, held Wednesday, May 19, 1886.*]

HALL OF THE INSTITUTE, May 19, 1886.

CHAS. H. BANES, President, in the Chair.

Present—144 members and four visitors.

The election of fourteen new members was reported.

Papers were read by MR. HOWARD M. DUBOIS on "Tests of Vehicle Wheels," and by MR. FRED. E. IVES, giving the results of some recent experiments on the preparation of "Color-Sensitive Plates for Photographic Purposes." Both papers, together with discussion thereon, have been referred for publication.

MR. PEDRO G. SALOM gave an oral account of the process of manufacturing castings of wrought iron and steel, called by the inventor, Mr. Peter Ostberg, of Stockholm, "Mitis" castings, and exhibited a number of specimens of the same.

Mr. Salom described the procedure to be substantially as follows:

Mr. Ostberg has made use of the well-known fact that certain alloys of metals possess a fusing temperature much lower than that of the metals composing them; and among these aluminium alloys are especially notable. In making "mitis" castings, a very small quantity, about $\frac{5}{100}$ of one per cent. of aluminium, in the form of a seven or eight per cent. aluminium alloy of cast iron, is added to the charge (about sixty pounds) of wrought iron in the crucible the moment this has been melted. The fusing point is at once lowered some 500°, and the charge, now an alloy of iron and aluminium, becomes extremely fluid and can be cast in the finest moulds, while the great difference between its temperature and its fusing point gives all the time necessary for manipulating it without danger of its solidifying. The extreme fluidity of the charge allows the ready escape of the gases, which otherwise would make a porous casting, and the result appears to be a remarkably fine, solid and tough casting of wrought iron.

The Secretary announced the death of two prominent members since last meeting, HENRY P. M. BERKENBINE and EMILE F. LOISEAU, and it was ordered that a special committee should be appointed by the President to prepare a suitable memorial of each of the deceased members.

The meeting thereupon proceeded to take action upon the report of the Committee on Reorganization, and was thereupon adjourned.

WM. H. WAHL, *Secretary.*

THE RELATIVE MERITS OF IRON AND COPPER WIRE FOR TELEGRAPH LINES.—Mr. W. H. Preece recently read a paper before the British Association, in which he discussed the results of experiments made to determine this question. His conclusions, which we give, indicate a wide market for copper:

"Copper is gradually replacing iron for aërial telegraphs, owing to its greater durability in the atmosphere; but its greater cost has led to the use of

smaller-sized wires. This can be done without detriment to the economy of the line, for the resistance of copper, as compared with iron, varies very nearly inversely as its price per ton, and hence the cost per mile remains about the same.

"It will be observed that copper shows a very decided superiority over iron, the speeds being as follows:

	Copper.	*Iron.*
Simplex working,	414	345 words per minute.
Duplex "	270	237 " " "

"It is anticipated that the superiority of copper over iron indicated by these experiments will have a beneficial and economical influence on our telegraph system, and that its extended use will enable us not only to work better, but to dispense with intermediate repeaters in many cases where, on long lines, they are now necessary.

"The most interesting point, however, in connection with these experiments is, that they apparently prove that the superiority of copper is not simply due to its smaller electrostatic capacity and resistance, but that it is more susceptible to rapid changes of electric currents than iron; for when the resistance and capacity of the copper and iron wires were equalized by the insertion of resistance coils and condensers, the speed on the former was not thereby diminished. Possibly the magnetic susceptibility of the iron is the cause of this. The magnetization of the iron acts as a kind of drag on the currents. It is well known that telephones always work better on copper than on iron wires, doubtless for the same reason.

"These experiments also show the high speed of working that is now attained by the post-office authorities with the Wheatstone automatic apparatus. The following table gives an interesting *résumé* of the different stages of the process made, and its rate of growth:

1877	80	words	per	minute.
1878	100	"	"	"
1879	130	"	"	"
1880	170	"	"	"
1881	190	"	"	"
1882	200	"	"	"
1883	250	"	"	"
1884	350	"	"	"
1885	420	"	"	"

—*Engineering and Mining Journal, Oct. 17, 1885.* C.

SPONTANEOUSLY REVERSIBLE LINES IN SPECTRA.—Cornu discusses various known facts, with special reference to the problem of deducing the distribution and relative intensities of the lines in the spectrum of an incandescent vapor from its chemical composition. He points out that lines which form periodic groups, are almost invariably spontaneously reversible. These lines close up toward the more refrangible end of the spectrum, and diminish in intensity. In the spectra of many metals the spontaneously reversible lines follow sensibly the same laws of distribution and intensity as the lines of hydrogen, and it would seem that this law of succession of the spectral lines, common to so many series, may possibly be expressed by the same function, which may be termed the hydrogenic function.—*Jour. Chem. Soc., Aug., 1885.* C.